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# RESULTS OF A STUDY OF THE EFFECT OF SPRAY VALVE RESIDUAL OIL ON DIESEL ENGINE GOVERNABILITY.

BY LIEUTENANT J. E. CANOOSE, U. S. NAVY, MEMBER.

Governors, when applied to the Diesel engine, retain their usual inherent faults. The usual question of centrifugal type versus inertia or combination types remains. Furthermore, if the governing system action is to be rapid, the governor must have sufficient power for rapid acceleration of the governor system linkage masses. Since the governor power varies as the square of its speed, for variable speed engines, and for large speed ranges, the maximum governor power must be very large if the minimum power is to be sufficient. Thus, a reliable governor may be condemned because

AUTHOR'S NOTE: Credit is due Machinist's Mate Harry P. Warren, U. S. Navy, for making all of the sketches in this article and for working out some of the diagrams.

run at a speed too low for the load it must accelerate. In general, the governor designer is blamed for a large part of poor Diesel engine governing for which he is not responsible.

Most Diesel engines of the air injection type have an inherent fault that makes them ungovernable in greater or lesser degree. This fault, which permits residual oil accumulation in spray valves, requires slight, if any ultimate sacrifice of simplicity for substantial reduction.

Governing here is intended to mean throttle governing, and, in particular, throttle governing for generator drive. This form of load, particularly where used for electric propulsion of ships, requires quick governing response. The demands of this service are extreme, since a motor may, during maneuvering, be driving a propeller at an overload rate and then be instantly relieved of all load. This means that the governing system of the prime mover must be sharply responsive in cutting off energy supply to the working cylinders if excessive rotative speeds are to be avoided without the use of flywheels of excessive mass. Further, since the prime mover is required to pick up large (zero to overload) instantaneous changes in load, it is essential that the governing system quickly reestablish the maximum rate of energy flow to the working cylinders. These two requirements are linked together so that an arrangement which satisfies one satisfies the other.

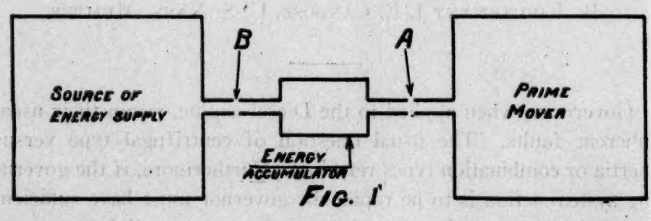


FIG. 1

Since it has been inferred that a governor which governs a steam engine well may fail on a Diesel engine of air injection type, it may be well to show more clearly what, in general, causes the difference.

Figure 1 is a diagrammatic sketch showing a source of energy supply, an energy accumulator, and a prime mover, in series. In

case of the steam cycle, the supply source is the steam boiler, the accumulator may be considered as the combined capacity of the superheater and steam line, and the prime mover is the reciprocating engine, or turbine. Points A and B indicate two possible places to fit a governor controlled valve. If the system were governed from point B it would obviously be unsatisfactory, if the accumulator capacity were appreciable, since the accumulator would drain after the governor had cut off the supply of steam. Moreover, the engine would not respond immediately when the steam flow were re-established since the accumulator must first be stored. Clearly, the governor valve must be as near to the driving elements of the engine as possible.

In the case of the air injection Diesel engine, good governing requires close-up throttling just as does the steam engine; but the energy storage capacity of spray valves is much larger than the corresponding storage capacity of the steam chest of the steam engine.

The spray valve residual oil capacity will now be considered. It is due to three basic factors:

(*a*) The type of spray valve.
(*b*) The characteristics of the fuel measuring pump.
(*c*) The fuel measuring pump timing.

Both the general type and detail characteristics of the spray valves under (*a*) have a marked influence on residual oil quantity.

## SPRAY VALVES.

Figure 2 shows the residual oil characteristics of the spray valve shown in Figure 3. The reduced areas shown were obtained by depressing a fuel pump suction valve by hand, thereby cutting off the supply of oil to the spray valve, after the full card had been taken, the indicator pencil being held to the drum through a few cycles succeeding the full card cycle to record the reduced areas. In every case above 200 H.P. secondary areas were recorded. In most cases tertiary areas were recorded, and in some cases quaternary areas appeared.

The fuel for producing the pressures shown by the reduced areas was drawn from the feeder passage in the side of the spray valve

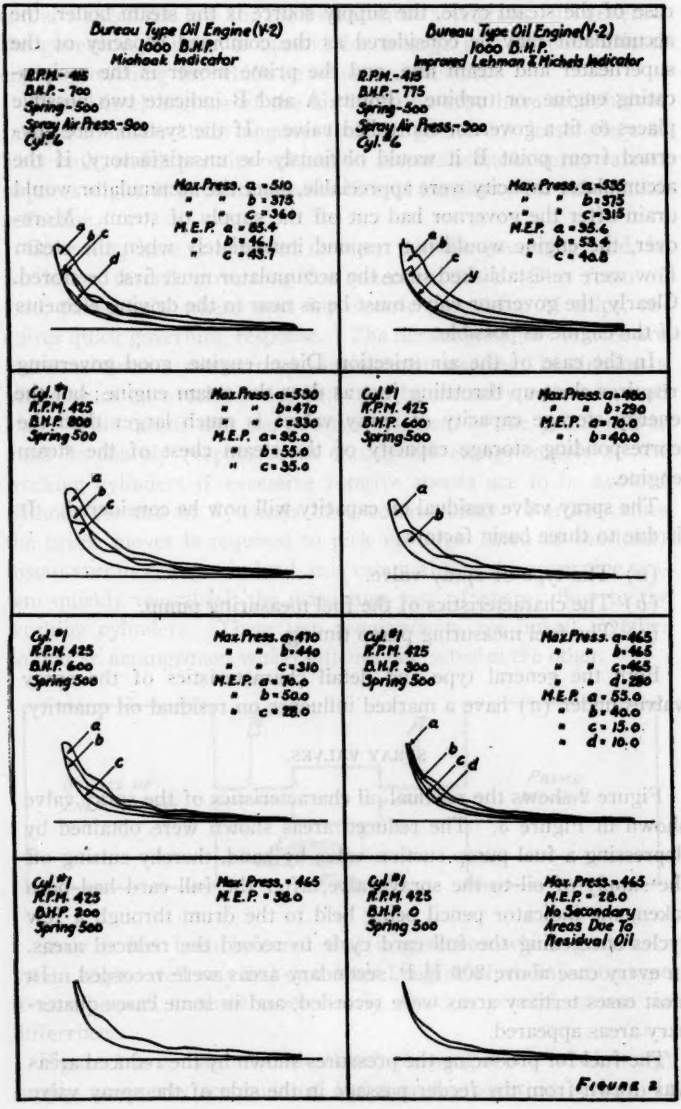

Bureau Type Oil Engine (Y-2)
1000 B.H.P.
Michaek Indicator
R.P.M.- 415
B.H.P.- 700
Spring - 500
Spray Air Press.- 900
Cyl. #6
Max. Press. a = 510
" " b = 375
" " c = 340
M.E.P. a = 85.4
" b = 14.1
" c = 43.7
a b c d
Bureau Type Oil Engine (Y-2)
1000 B.H.P.
Improved Lehman & Michels Indicator
R.P.M.- 418
B.H.P.- 775
Spring - 500
Spray Air Press.- 900
Cyl. #6
Max. Press. a = 535
" " b = 375
" " c = 315
M.E.P. a = 35.4
" b = 53.4
" c = 40.8
a b c y
Cyl. #1
R.P.M. 425
B.H.P. 800
Spring 500
Max. Press. a = 530
" " b = 470
" " c = 330
M.E.P. a = 95.0
" b = 55.0
" c = 35.0
a b c
Cyl. #1
R.P.M. 425
B.H.P. 600
Spring 500
Max. Press. a = 480
" " b = 290
M.E.P. a = 70.0
" b = 40.0
a b
Cyl. #1
R.P.M. 425
B.H.P. 600
Spring 500
Max. Press. a = 470
" " b = 440
" " c = 310
M.E.P. a = 70.0
" b = 50.0
" c = 28.0
a b c
Cyl. #1
R.P.M. 425
B.H.P. 300
Spring 500
Max. Press. a = 465
" " b = 465
" " c = 465
" " d = 280
M.E.P. a = 55.0
" b = 40.0
" c = 15.0
" d = 10.0
a b c d
Cyl. #1
R.P.M. 425
B.H.P. 200
Spring 500
Max. Press. = 465
M.E.P. = 38.0
Cyl. #1
R.P.M. 425
B.H.P. 0
Spring 500
Max. Press. = 465
M.E.P. = 28.0
No Secondary
Areas Due To
Residual Oil

FIGURE 2

body and from the annulus in the atomizer which this feeder port supplies. It is to be noted that the secondary areas are in each case more than 50 per cent of the full area. If, then, the friction horsepower of the engine be assumed to be 25 per cent, a substantial margin of power is available for acceleration during the following one or two cycles.

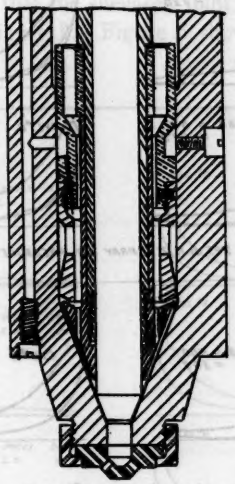


FIGURE 3

The following table gives the secondary and tertiary mean effective pressures and their values as percentages of 95 pounds per square inch. This M.E.P. value being that for 800 H.P. 425 R.P.M.

| B.H.P. | R.P.M. | Full M.E.P. | Secondary M.E.P. | Secondary Percentage | Tertiary M.E.P. | Tertiary Percentage |
|---|---|---|---|---|---|---|
| 800 | 425 | 95 | 55 | 58 | 35 | 36.9 |
| 775 | 415 | 95.6 | 53.4 | 56.2 | 40.8 | 43.0 |
| 700 | 415 | 85.4 | 46.1 | 48.5 | 43.7 | 46.0 |
| 600 | 425 | 70 | 50 | 52.6 | 28 | 29.5 |
| 600 | 425 | 70 | 40 | 42.2 | none | 00.0 |
| 300 | 425 | 55 | 40 | 42.2 | 15 | 15.8 |
| 200 | 425 | 38 | 00.0 | 00.0 | 00.0 | 00.0 |
| Idling | 425 | 28 | 00.0 | 00.0 | 00.0 | 00.0 |

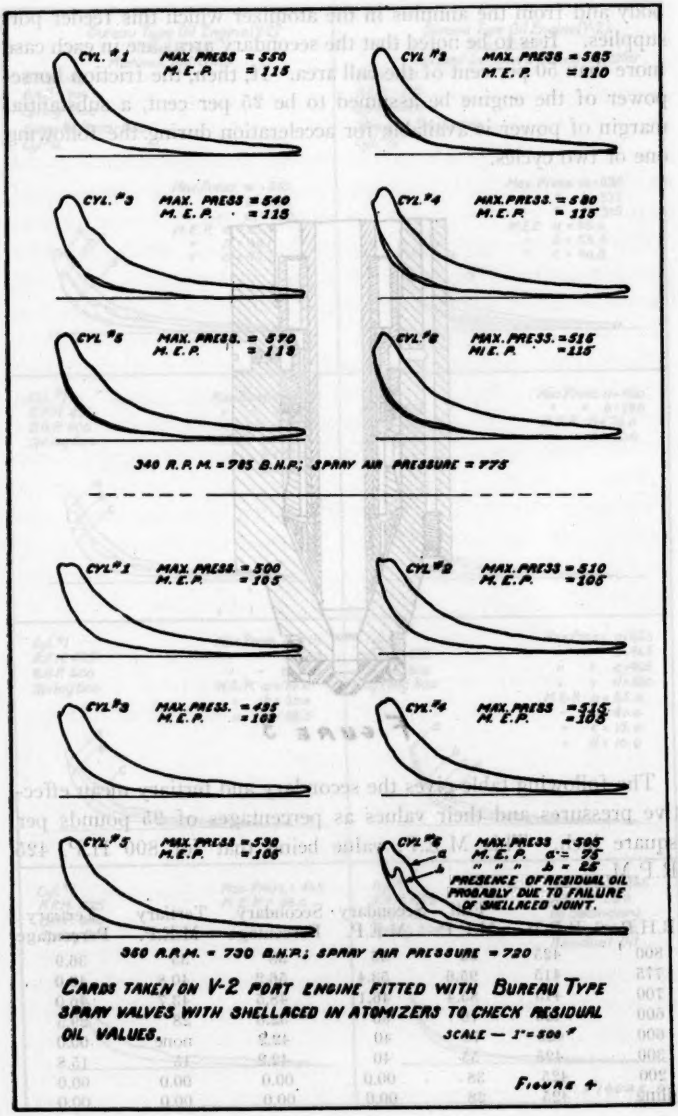
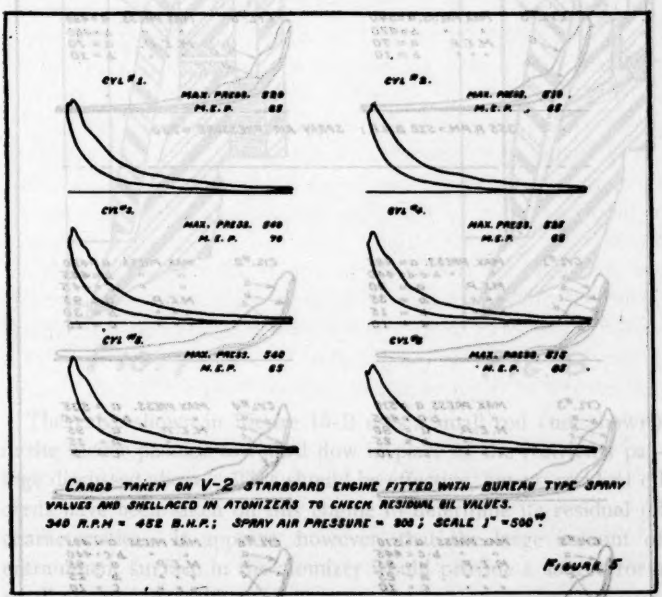

CYL. #1 MAX. PRESS = 550 M.E.P. = 115
CYL #2 MAX. PRESS = 585 M.E.P. = 110
CYL. #3 MAX. PRESS = 540 M.E.P. = 115
CYL #4 MAX. PRESS. = 580 M.E.P. = 115
CYL #5 MAX. PRESS. = 570 M.E.P. = 110
CYL #6 MAX. PRESS. = 516 M.E.P. = 115
340 R.P.M. = 785 B.H.P.; SPRAY AIR PRESSURE = 775
CYL #1 MAX. PRESS. = 500 M.E.P. = 105
CYL #2 MAX. PRESS = 510 M.E.P. = 105
CYL #3 MAX. PRESS. = 495 M.E.P. = 105
CYL #4 MAX. PRESS = 515 M.E.P. = 105
CYL #5 MAX. PRESS = 530 M.E.P. = 105
CYL #6 MAX. PRESS = 595
a M.E.P. a = 75
" " " b = 25
PRESENCE OF RESIDUAL OIL PROBABLY DUE TO FAILURE OF SHELLACED JOINT.
360 R.P.M. = 730 B.H.P.; SPRAY AIR PRESSURE = 720
CARDS TAKEN ON V-2 PORT ENGINE FITTED WITH BUREAU TYPE SPRAY VALVES WITH SHELLACED IN ATOMIZERS TO CHECK RESIDUAL OIL VALUES.
SCALE — 1" = 500 #

FIGURE 4

The residual oil characteristic of this valve is greatly influenced by the tightness of fit between the atomizer element and the valve body. The residual oil areas shown in diagrams of Figure 2 are due to loose atomizer element fits in the valve body. This was proved* by setting the atomizers in stiff shellac, thus sealing the joints. The result of this procedure is shown in Figure 4. Here the secondary areas are very small, where present, except in one case where it appears that the shellaced joint failed, giving a card similar to those of Figure 2. Figure 5 shows effect of reduced power.

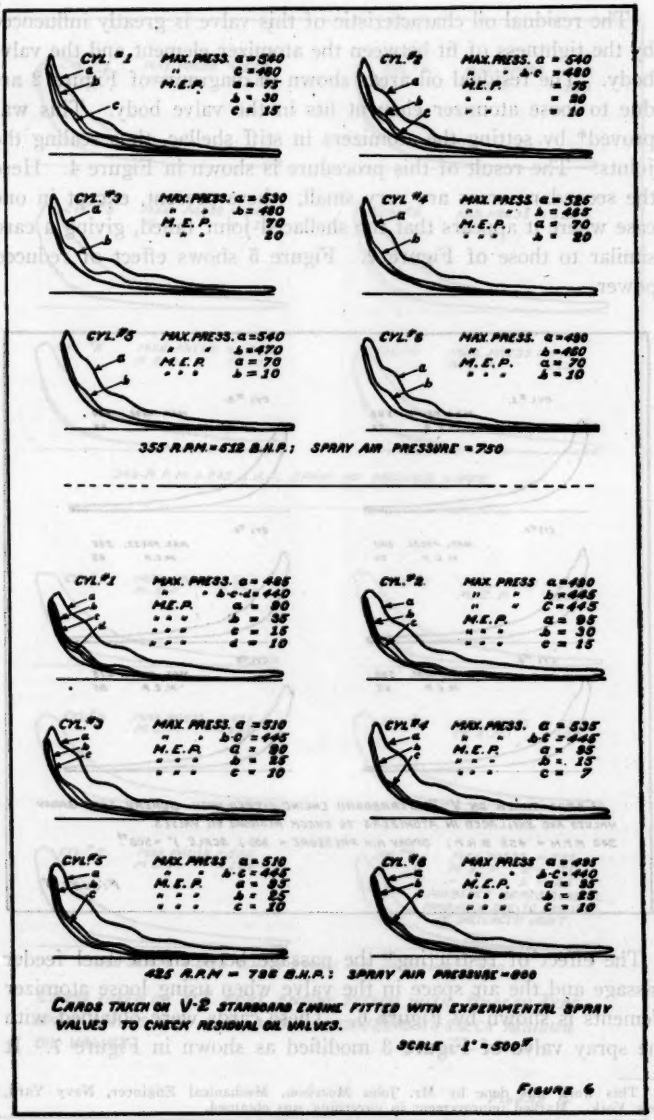



CARDS TAKEN ON V-2 STARBOARD ENGINE FITTED WITH BUREAU TYPE SPRAY VALVES AND SHELLACED IN ATOMIZERS TO CHECK RESIDUAL OIL VALVES.
340 R.P.M = 452 B.H.P.; SPRAY AIR PRESSURE = 900; SCALE 1" = 500#.

FIGURE 5

The effect of restricting* the passage between the fuel feeder passage and the air space in the valve when using loose atomizer elements is shown by Figure 6. These cards were obtained with the spray valve of Figure 3 modified as shown in Figure 7. It

* This work was done by Mr. John Morrison, Mechanical Engineer, Navy Yard, New York. Marked improvement in governing was obtained.




CARDS TAKEN ON V-2 STARBOARD ENGINE FITTED WITH EXPERIMENTAL SPRAY VALVES TO CHECK RESIDUAL OIL VALVES.

SCALE 1" = 500#

FIGURE 6

will be noted here that the residual oil areas are much smaller than in the case of the original valve. If the passage is still further restricted as shown in Figure 8, no secondary areas occur. Figure 8 is a section of the spray valve body used in the Bureau type 450 H.P. engine.

FIG. 7

FIG. 8

The valve shown in Figure 15-B uses a small rod (not shown) in the feeder passage to retard flow in place of the restricted passage discussed above. This should be effective, but no residual oil cards have been taken on this engine to determine its residual oil characteristics. It appears, however, that the large amount of entrainment surface in the atomizer would provide a source for a small amount of residual oil.

So far as is known to the writer, no data on residual oil characteristics of any spray valve other than the Bureau type have been published.

In light of experiments that have been run, there is no doubt that small differences in design, or in workmanship of manufacture, may make large differences in the residual oil characteristics

of a spray valve. Further, design features favorable to residual oil elimination may be adverse to proper atomization, though both types of Bureau valves are satisfactory in both these respects if properly manufactured and assembled.

### FUEL PUMP.

The fuel measuring pump characteristics which affect residual oil values are:

(1) The number of plungers driven in one group.
(2) The type of control valve used.
(3) The full power delivery angle used.

(1) The effect, on residual oil values, of grouping plungers is obvious. The greater the number of plungers in one group, the greater the number of charges to be burned out after governor action; hence, the greater the overspeeding due to this cause.

(2) Some fuel measuring pumps use a controlled bypassing valve of piston type for regulating the amount of fuel delivered to the spray valve. Other than friction, this valve offers no reaction to the governor action. With this arrangement the governor can cut off the fuel supply to a spray valve at any point in the pump cycle; which means it may stop pump delivery *during discharge* to the spray valve. Therefore, no residual oil accumulates in spray valves due to pump action *after* governor action. This non-reactive valve offers better governor regulation than the reactive type.

The controlled suction valve is used in all U. S. Navy engines. This valve cannot be opened by governor action during a discharge stroke since pressures ranging from 900 to 1800 pounds per square inch hold it seated. Therefore, where reactionary type fuel pump control valves are used, the governor can act only between discharge strokes. In this case, therefore, residual oil *does* accumulate due to pump action *after* governor action.

(3) The effect of changing the fuel pump full power delivery angle (using controlled suction valve) is very marked. In general, large delivery angles mean large residual oil values, and vice versa. The limiting residual oil minimum occurs in the case where the delivery angle equals, and is coincident with, the injection angle.

This obtains in the solid injection engine and accounts for the fact that solid injection engines are more easily governed than air injection engines. The limiting maximum occurs where the pump effective delivery angle is 180 degrees of pump angle and where pump runs at one-half crank-shaft speed. The pump then delivers through 360 degrees of working crank angle.*

Figure 9 shows the effect of changing the delivery angle from that equal to, and coincident with, the injection angle, to 360 degrees of working crank angle with the end of fueling angle coincident with the beginning of injection. In both cases reactionary type suction valves are assumed. As the figure shows, the difference due to this change is exactly three full charges, or 50 per cent of a total engine charge.

Figure 10 shows the effect of changing the full power delivery angle from the usual 90 degrees of pump crank angle to 60 degrees of pump crank angle. The maxima are reduced 10 per cent, but the average is greatly reduced.

## FUEL PUMP TIMING.

The effect of changing the fuel pump timing is also great. As shown above, the ideal case obtains where the fuel pump full power delivery angle is equal to the injection angle and is coincident therewith. This, however, involves the question of fueling the spray valve while injection is in progress, and will be discussed in detail below.

Figure 11 shows the effect of shifting the delivery arc from ahead of the injection angle to the position where delivery starts at the end of injection. Miniature diagrams at the left of the curves delineate this change. The result of this change is to increase the residual oil maxima from two to six full charges, or 67 per cent of a full engine charge. The duration of the maxima is also increased.

The proper angular position for the pump delivery arc is dependent on the spray valve characteristics and on the fuel pump speed. Study of German designs indicates that they particularly avoided fuelling spray valves during the injection period. This is

* It should be noted that this refers only to the delivery angle which effect is distinct from that of change in timing.

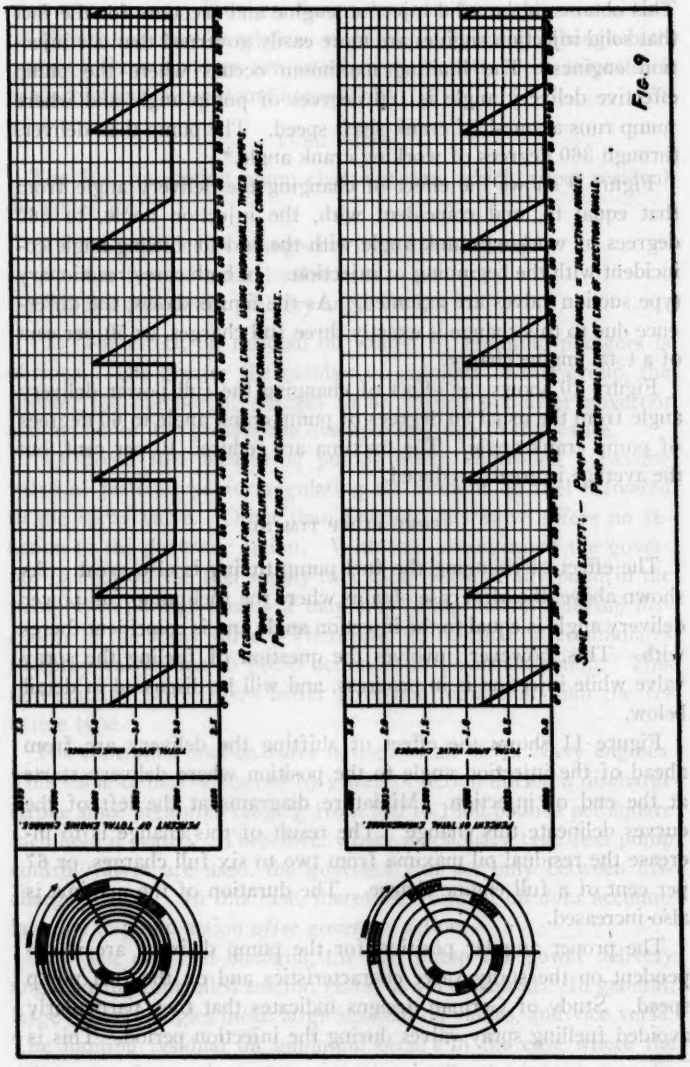


FIG. 9

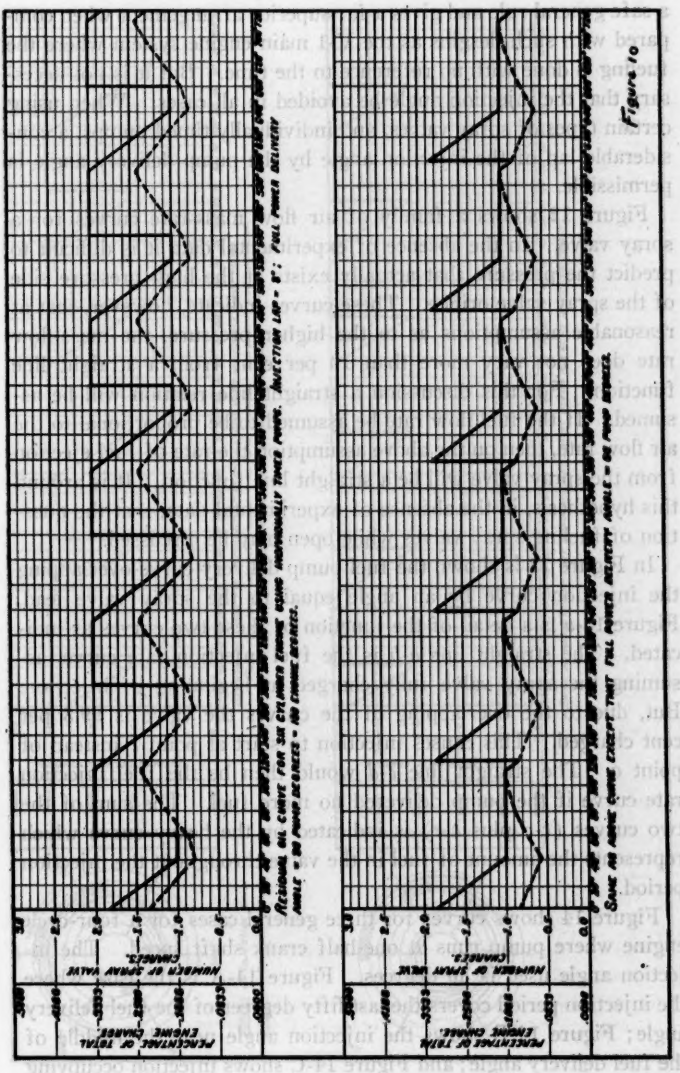

PERCENTAGE OF TOTAL ENGINE CHARGE.
NUMBER OF SPRAY VALVE CHARGES.
RECORDING OIL CURVE FOR SIX CYLINDER ENGINE USING INDIVIDUALLY TIMED PUMPS. INJECTION LAG = 0°; FULL POWER DELIVERY ANGLE = 20 PUMP DEGREES = 120 ENGINE DEGREES.
PERCENTAGE OF TOTAL ENGINE CHARGE.
NUMBER OF SPRAY VALVE CHARGES.
SAME AS ABOVE CURVE EXCEPT THAT FULL POWER DELIVERY ANGLE = 60 PUMP DEGREES.

Figure 10

a safe general rule and gives a far superior arrangement when compared with such desgins as the V-1 main engine system where the fueling is done with no reference to the time. But it is not necessary that the injection angle be avoided in all cases. When using certain types of spray valves, and individually timed pumps, a considerable lap of the injection angle by the pump delivery angle is permissible.

Figure 12 shows a family of air flow mass-rate curves for a spray valve. In the absence of experimental data it is difficult to predict the pressure that actually exists at the high pressure side of the spray valve orifice. These curves indicate, however, that at reasonable assumptions as to the higher pressure, the mass-flow rate does not vary more than 10 per cent from a straight line function. For this discussion a straight line relation will be assumed. If the fuel flow rate be assumed to be proportional to the air flow rate, then on the above assumption the rate of fuel ejection from the spray valve will be a straight line function. It is around this hypothesis, in the absence of experimental data, that the question of fueling spray valves while open may be discussed.

In Figure 13 is shown the fuel pump delivery curve overlapping the injection curve by an angle equal to the spray valve lead. Figure 13-*a* is a detail of the junction of these two curves as indicated. The straight line *a-b* is the fuel injection rate curve, assuming the spray valve fully charged at beginning of injection. But, due to the overlapping of the curves the valve is 99.3 per cent charged. This causes injection to start at point *c* instead of point *a*. The straight line *c-d* would then be the fuel injection rate curve if the pump delivered no more fuel. The sum of the two curves (*c-d* plus *c-e*) is indicated by the heavy curve which represents the amount of fuel in the valve throughout the injection period.

Figure 14 shows curves for three general cases for a four-cycle engine where pump runs at one-half crank shaft speed. The injection angle used is 50 degrees. Figure 14-A is the case where the injection period covers the last fifty degrees of the fuel delivery angle; Figure 14-B shows the injection angle near the middle of the fuel delivery angle; and Figure 14-C shows injection occupying the first fifty degrees of fueling angle. These cases merely indi-

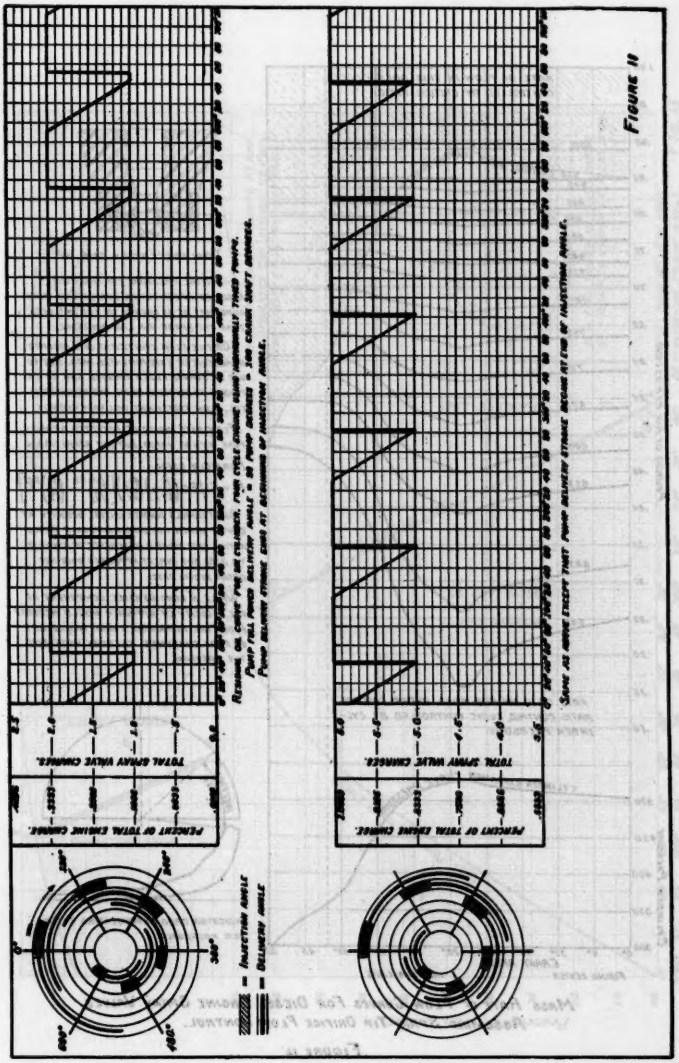
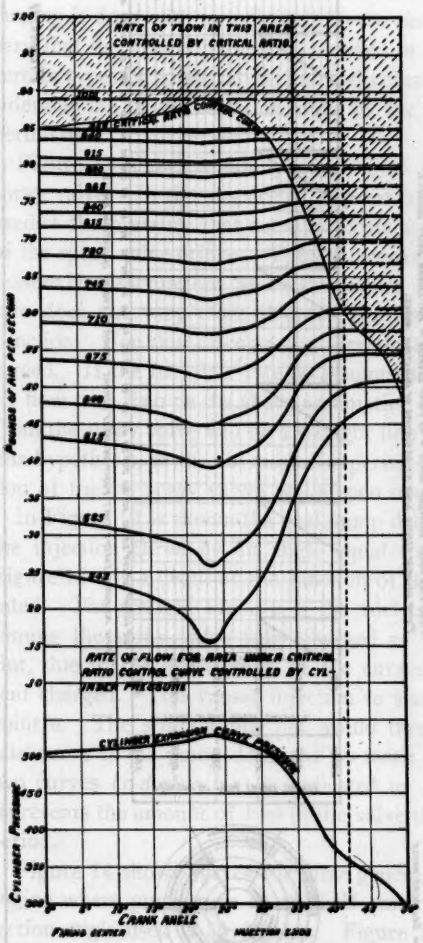

PERCENT OF TOTAL ENGINE CHARGE.
TOTAL SPRAY VALVE CHARGE.
= INJECTION ANGLE
= DELIVERY ANGLE

FIGURE 11



TOTAL ORIFICE AREA = .0436 in².

INITIAL PRESSURE FOR THIS POINT ASSUMED IN COMPUTING CURVES. THIS PRESSURE MAY BE AS MUCH AS 100 LBS IN² LESS THAN THE SPRAY AIR LINE PRESSURE.

ORIFICE FLOW COEFFICIENT OF 1 ASSUMED IN ABSENCE OF EXPERIMENTAL DATA. DUE TO USE OF SHARP EDGED ORIFICE AND TO EXPOSING TO PRESSURE ABOVE AND BELOW CRITICAL, FLOW COEFFICIENT WILL VARY SLIGHTLY.

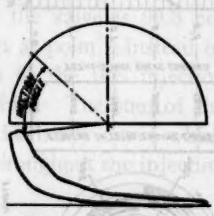
FLOW RATE WOULD BE AFFECTED BY MIXTURE WITH OIL VAPOR, BUT NOT APPRECIABLY.

FORMULA USED:-

$$M = F\sqrt{2g\left(\frac{K}{K-1}\right)(G_1)\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{K}} - \left(\frac{P_2}{P_1}\right)^{\frac{K+1}{K}}\right]}$$

THIS FORMULA IGNORES INITIAL VELOCITY OF ORIFICE WHICH FOR DIRECT APPROACH WOULD IN THIS CASE MAKE A DIFFERENCE OF ABOUT 7 TO 8%. BUT DIRECT APPROACH DOES NOT OBTAIN IN THIS CASE.

WHILE A CONSIDERABLE COMPONENT OF THE VELOCITY OF APPROACH WOULD INCREASE FLOW, THE REMAINING ENERGY WOULD SET UP EDDY CURRENTS THEREBY INCREASING THE PRESSURE.




INDICATOR CARD REPRESENTING CYLINDER PRESSURES.

*Mass Rate of Flow Curves For Diesel Engine Spray Valves Assuming Spray Tip Orifice Flow Control.*

FIGURE 18

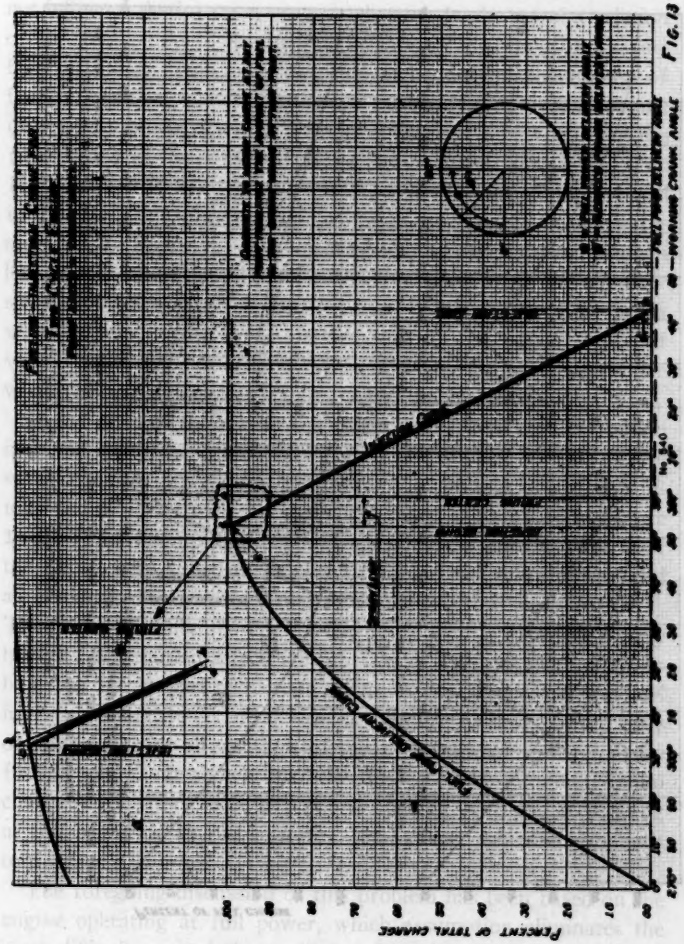


FIG. 13

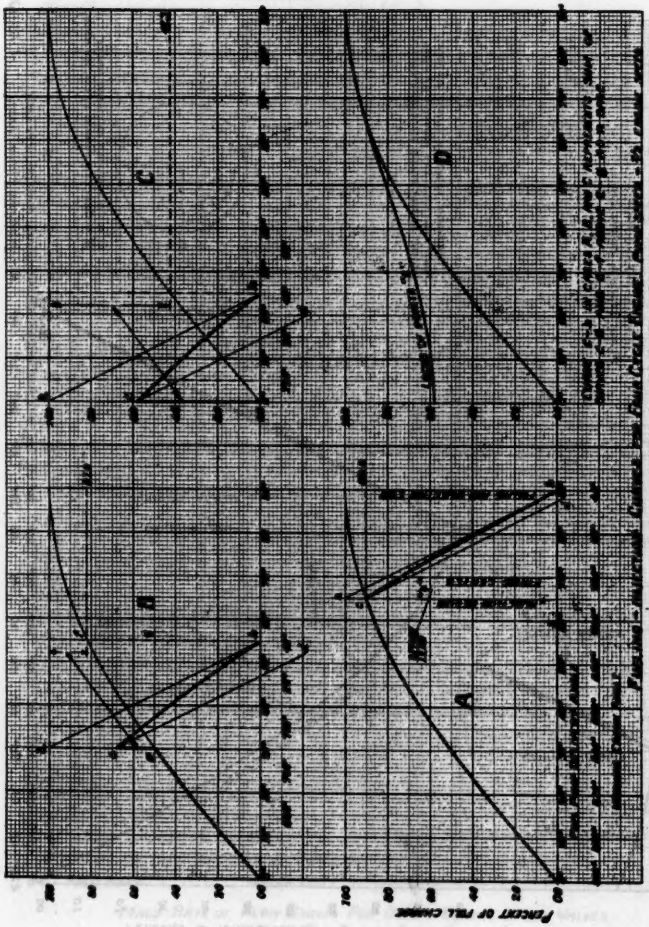


FIG. 14

cate that if the rate of fuel flow is proportional to the rate of air flow, and if the amount of fuel in the valve does not affect this flow relation, then the rate of fuel delivery to the spray valve will not influence the injection process whatever be the angular relation of the fuel pump to the injection angle. However, the air-fuel flow relation in some valves is clearly dependent on the amount of fuel in the valve. The valve for the Bureau type 1000 H.P. engine (Figure 3) and the valve of Figure 15-B are of this type. But the function of the valves used in the Bureau type 3000 H.P., 1750 H.P. and 450 H.P. engines is less dependent on the amount of fuel in the valve, since the fuel is delivered to the space from which it is ejected as in Figure 8. This ejecting action, after starting, is much less dependent on the oil head in the annulus between the atomizer sleeve and the valve body, than it is on the unbalanced air heads which obtain during flow. In the case of the former types of valves, however, the air must pass through the oil in the valve whatever be the portion of a full charge in the valve.

In the cases of the valves of Figures 3 and 15-B the distribution it seems would be greatly affected by the amount of fuel in the valve when air flow starts. For example, assume an engine running at full power with the fuel pump timed as indicated in Figure 14-B. Then the initial distribution would be about the same as if the pump were timed to deliver all fuel ahead of injection, and as if the engine were running at two-thirds full indicated H.P. The fuel represented by G-F of the figure would, therefore, surely have its distribution affected by the pump delivery rate. It is to be noted that at start of injection there is only 10 per cent less fuel in the valve in case C than in case B. Figure 14-D shows the locus of points "C" for all positions of the injection angle relative to the fueling angle. It shows, that for this case, the minimum initial charge in the spray valve at point of opening, with engine running at full power, is 58 per cent of a total charge which corresponds to the case of Figure 14-C.

The foregoing discussion of this problem has been based on the engine operating at full power, which assumption eliminates the most difficult part of the problem. This is the question of providing, at reduced powers, an initial or "pilot" charge of fuel sufficient to anticipate blast chilling and insure ignition in cool cylin-

ders. It is not desirable to fuel a spray valve immediately after it closed because a leaking valve so fueled continues injecting fuel during the expansion angle thereby causing serious burning of exhaust valves and ports. This, then, forbids the angular relations between pump and spray valve as indicated in Figures 14-B and 14-C. It is interesting to note, however, that the relation 14-B would assure a pilot charge of 18.1 per cent of the full charge with engine running at minimum power. Also, the relation shown in 14-C would put all the reduced charge in the valve immediately following the end of injection when reduced power delivery angles below 65 degrees were being used. The relation of Figure 14-A remains for consideration.

Figure 15 shows four possible cases and indicates the effect of each. All are in terms of pump crank angle, where pump runs at half engine speed.

Case 1.—The fuel pump delivery center coincides with the end of injection. A 90-degree full power delivery angle is used. The injection angle used is 58 degrees, which is 29 degrees of pump crank angle. As shown, 12.5 per cent of the full charge is delivered to the valve after injection starts, and a 12 per cent * pilot charge would set 24.5 per cent of a full charge as the minimum throttle setting for the engine. This would make satisfactory idling for the engine questionable.

Case 2.—This case is the same as case 1 except that a 60-degree full power delivery angle is assumed. This results in 25.1 per cent of a full charge being delivered after the start of injection, and a minimum indicated H.P. of 37.1 per cent to provide a pilot charge of 12 per cent. This would obviously be unsatisfactory for low powers.

Case 3.—Here the fuel pump angle laps one-half of the injection angle. Otherwise it is the same as case 1. The results are 3.2 per cent of a full charge delivered after injection starts, and a 15.2 per cent minimum I.H.P. requirement to allow 12 per cent pilot charge. This would obviously be a satisfactory minimum power requirement.

Case 4.—This is the same as case 2 except that the fuel pump angle laps one-half of the injection angle as in case 3. The

* Twelve per cent is a liberal figure.

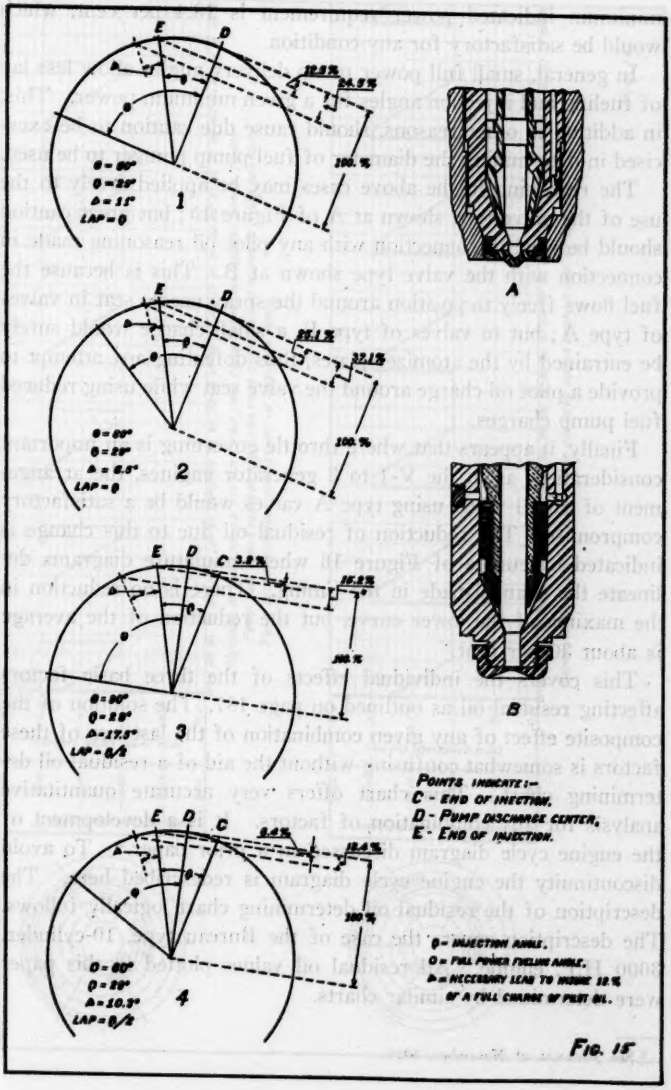

E
D
C
A
B
θ = 80°
θ = 25°
Δ = 1.5°
LAP = 0
1
12.5%
24.5%
100 %
θ = 80°
θ = 25°
Δ = 6.5°
LAP = 0
2
25.1%
32.1%
100 %
θ = 80°
θ = 25°
Δ = 17.5°
LAP = θ/2
3
3.2%
16.2%
100 %
θ = 80°
θ = 25°
Δ = 10.3°
LAP = θ/2
4
6.4%
19.4%
100 %
A
B
POINTS INDICATE:—
C - END OF INJECTION.
D - PUMP DISCHARGE CENTER.
E - END OF INJECTION.
θ = INJECTION ANGLE.
Θ = FULL POWER YIELDING ANGLE.
Δ = NECESSARY LEAD TO INSURE 12 %
OF A FULL CHARGE OF PILOT OIL.

FIG. 15

minimum indicated power requirement is 18.4 per cent, which would be satisfactory for any condition.

In general, small full power pump delivery angles allow less lap of fueling and injection angles for a given minimum power. This, in addition to other reasons, should cause due caution to be exercised in determining the diameter of fuel pump plunger to be used.

The reasoning in the above cases may be applied freely to the use of the valve type shown at A of Figure 15; but great caution should be used in connection with any pilot oil reasoning made in connection with the valve type shown at B. This is because the fuel flows freely to position around the spray needle seat in valves of type A; but in valves of type B, a small charge would surely be entrained by the atomizer plates, thus defeating any attempt to provide a pilot oil charge around the valve seat while using reduced fuel pump charges.

Finally, it appears that where throttle governing is an important consideration, as in the V-1 to 3 generator engines, the arrangement of case 1 when using type A valves would be a satisfactory compromise. The reduction of residual oil due to this change is indicated by curves of Figure 16 where miniature diagrams delineate the change made in the timing. There is no reduction in the maxima of the lower curve, but the reduction of the average is about 30 per cent.

This covers the individual effects of the three basic factors affecting residual oil as outlined on page 187. The solution of the composite effect of any given combination of the last two of these factors is somewhat confusing without the aid of a residual oil determining chart. This chart offers very accurate quantitative analysis for any combination of factors. It is a development of the engine cycle diagram discussed in a prior paper.* To avoid discontinuity the engine cycle diagram is redescribed here. The description of the residual oil determining chart logically follows. The description covers the case of the Bureau type, 10-cylinder, 3000 H.P. engine. All residual oil values plotted in this paper were determined by similar charts.

* See JOURNAL of November, 1924.

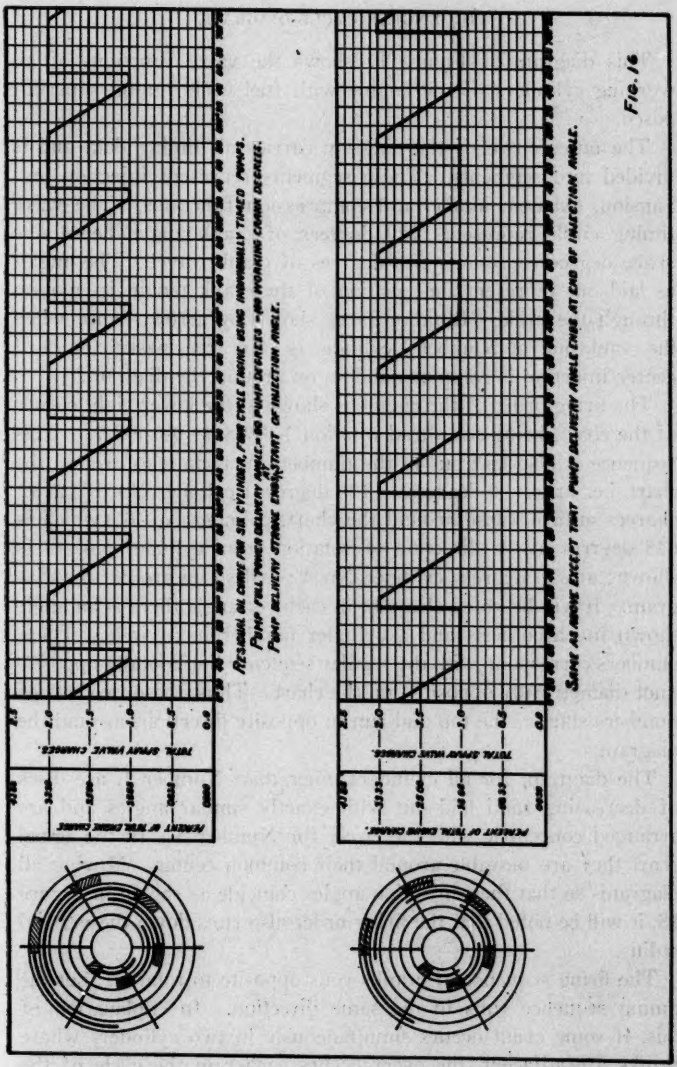

RESIDUAL OIL CURVE FOR SIX CYLINDER, FOUR CYCLE ENGINE USING INDIVIDUALLY TIMED PUMPS.
PUMP FULL POWER DELIVERY 30 DEG.=60 PUMP DEGREES =120 WORKING CRANK DEGREES.
PUMP DELIVERY STROKE ENDS START OF INJECTION ANGLE.
PERCENT OF TOTAL CHARGE.
TOTAL SPRAY VALVE CHARGES.
SAME AS ABOVE EXCEPT THAT PUMP DELIVERY ANGLE ENDS AT END OF INJECTION ANGLE.

FIG. 16

## THE ENGINE CYCLE DIAGRAM.

This diagram of Figure 17 shows the valve functions of all working cylinders of the engine with fuel cycle timing superimposed.

The outer circle of the diagram carries all timing data and is divided into segments. These segments represent injection, expansion, exhaust, intake, and compression functions. The valve timing circle represents 720 degrees of crank angle, hence one space degree represents two degrees of crank travel. The angles as laid out represent the position of the crank pin in its motion through the cycle. Rotation is as shown by arrow, hence when the crank pin for Number 1 cylinder is 44 degrees past firing dead center injection is complete, and so on around the diagram.

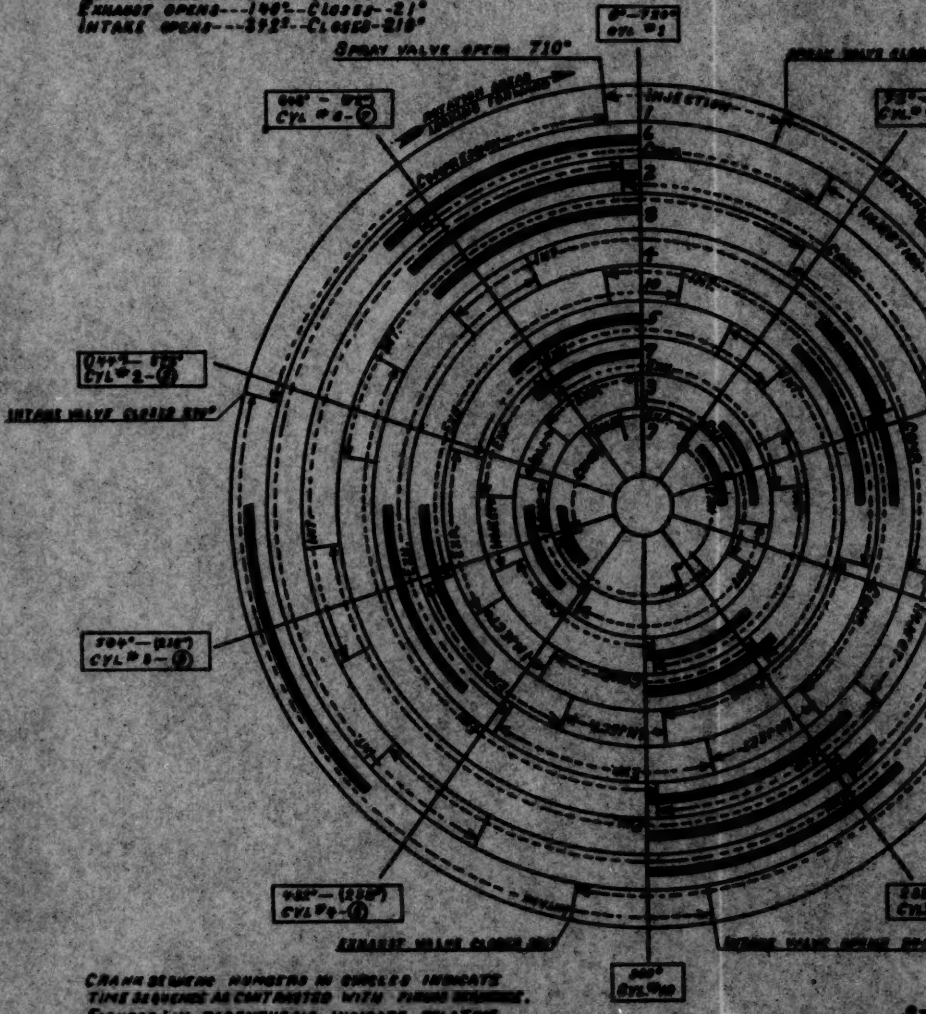
The firing order of the engine is shown at the lower right corner of the chart, being for ahead rotation 1-6-2-8-4-10-5-9-3-7. This sequence is also laid out in the numbers in rectangles around the chart, i.e., at top "0 degrees—(720 degrees) cylinder No. 1"; at 72 degrees anticlockwise around the chart, cylinder No. 6 (which is 648 degrees in the direction of rotation from cylinder No. 1) is shown; at 576 degrees cylinder No. 2; and so on around the diagram. It will be noted also that in each rectangle there is an angle shown in parenthesis and a cylinder number in a circle. These numbers correspond with the regular sequence horizontally opposite (not diametrically opposite) on the chart. Thus, these two sets of numbers start at the top and run in opposite directions around the diagram.

The diagram, for all cylinders other than Number 1, are discs of decreasing radii laid out with exactly similar angles and are arranged concentric with the circle for Number 1. In the actual chart they are movable around their common center. Moving all diagrams so that their injection angles coincide as shown in Figure 18, it will be noted that the firing order also runs down the vertical radius.

The firing sequence of cranks runs opposite to rotation; but the timing sequence runs in the same direction. In explanation of this, if some event occurs simultaneously in two cylinders whose cranks are adjacent, the event occurs earlier in the cycle of the following crank than in the cycle of the leading crank. As shown

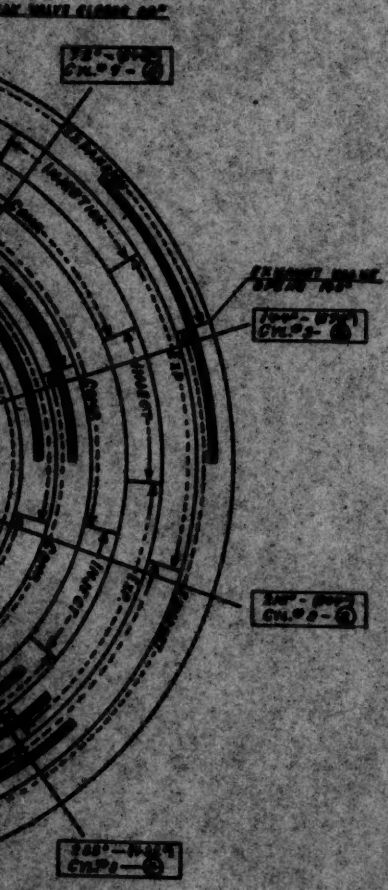
VALVE SETTING
SPRAY OPENS----355°--CLOSES--42°
EXHAUST OPENS---140°--CLOSES--21°
INTAKE OPENS---332°--CLOSES-210°

FUEL EFFICIENT CYCLE
TYPE DIESEL ENGI




CRANK STROKE NUMBERS IN CIRCLES INDICATE TIME SEQUENCE AS CONTRASTED WITH FIRING SEQUENCE. FIGURES IN PARENTHESIS INDICATE RELATIVE PORTIONS OF CRANKS IN THEIR TIME SEQUENCE IN DEGREES.

Engine Cycle of 3000 H.P. Burean
Diesel Engine.



Starboard Engine

VALVE SETTINGS
SPRAY OPENS---355°---CLOSES---42°
EXHAUST OPENS---140°---CLOSES---21°
INTAKE OPENS---342°---CLOSES---210°
TWO DEGREES OF CRANK TRAVEL = ONE DEGREE IN DIAGRAM.

FUEL ENGINE CY
DIESEL ENGINE.

0°—720°
CYL. #1

SPRAY VALVE OPENS 710°

SPRAY VALV

848°—(128°)
CYL. #6—⑦

576°—(144°)
CYL. #2—③

INTAKE VALVE
CLOSES 570°

504°—(216°)
CYL. #8—⑤

432°—(288°)
CYL. #4—⑤

EXHAUST VALVE CLOSES 381°

380°
CYL. #10

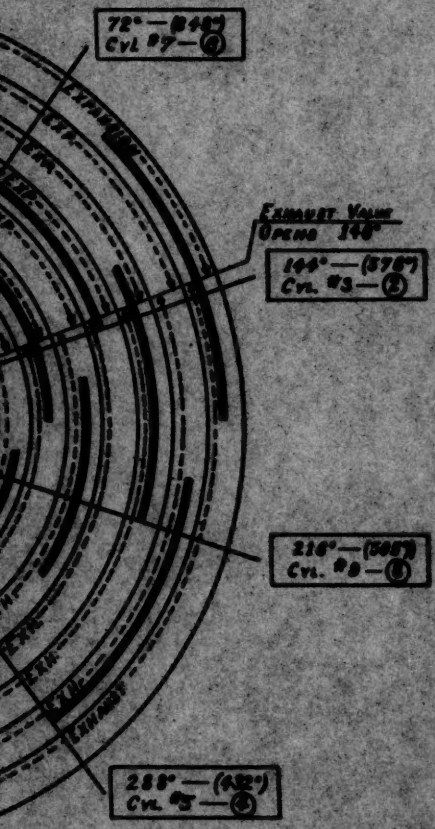
INTAKE VALVE OPE

CRANK SEQUENCE NUMBERS IN CIRCLES INDICATE TIME SEQUENCE AS CONTRASTED WITH FIRING SEQUENCE. FIGURES IN PARENTHESIS INDICATE RELATIVE POSITIONS OF CRANKS.— THEIR TIME SEQUENCE IN DEGREES.

FIRING ORDER—(1
FIRING ORDER—(A

...L ENGINE CYCLE or 3000 H.P. BUREAU TYPE
...EL ENGINE.

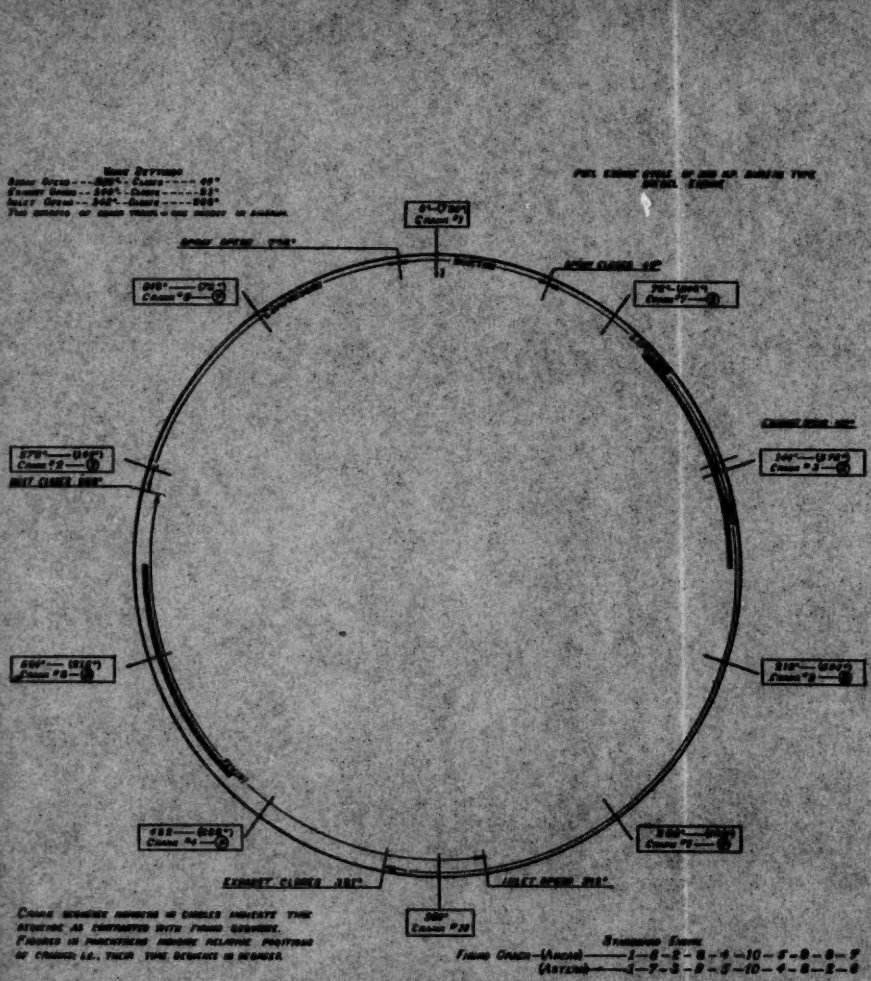
...RAY VALVE CLOSES 42°

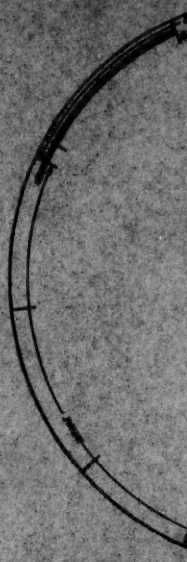


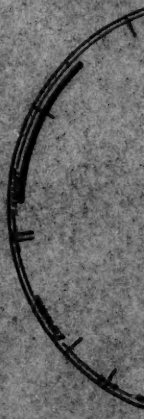

...VALVE OPENS 348°

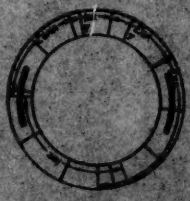
...ORDER — (FIRING) - - - 1-6-2-5-4-10-5-9-3-7
...ORDER — (ANTERN) - - 1-7-3-9-5-10-4-8-2-6

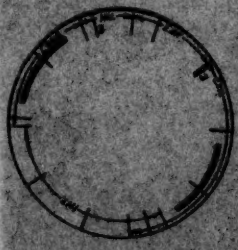
FIGURE 18











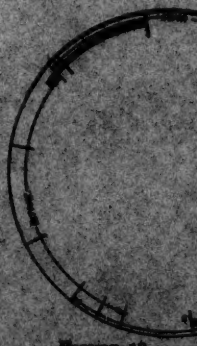
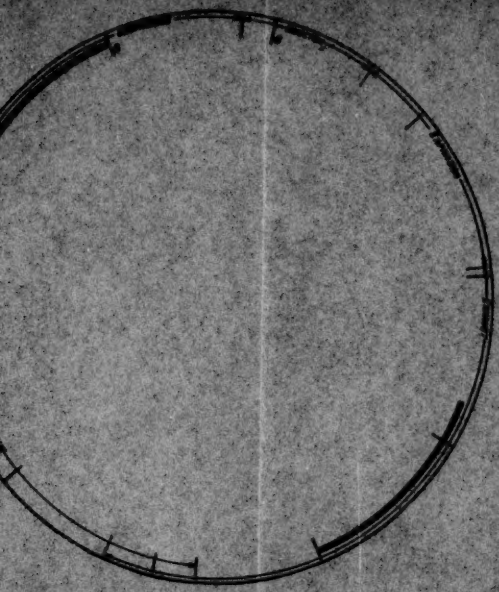
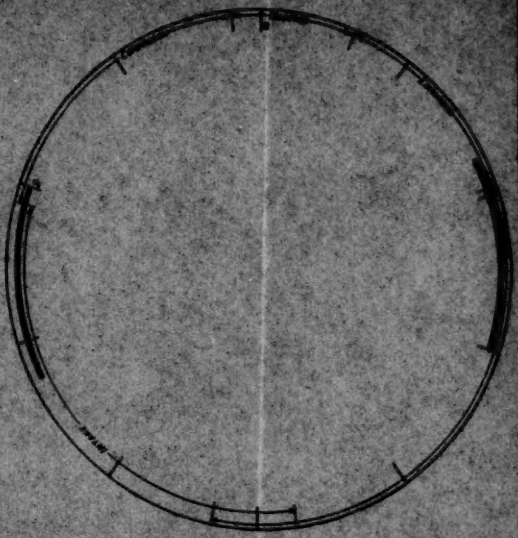
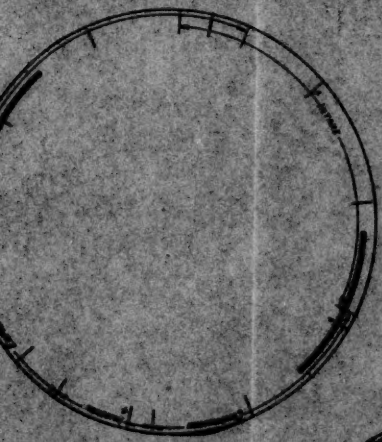
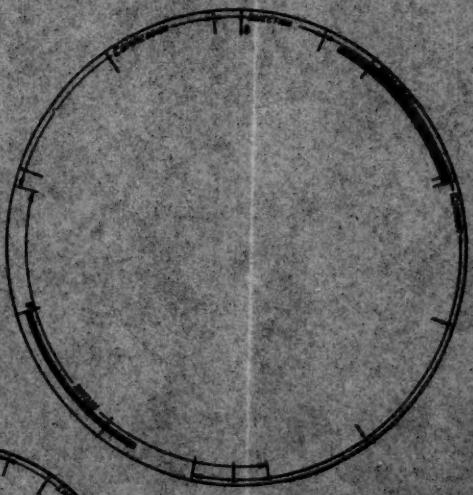


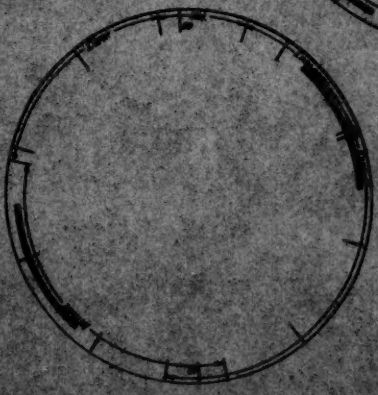
FIGURE 19.

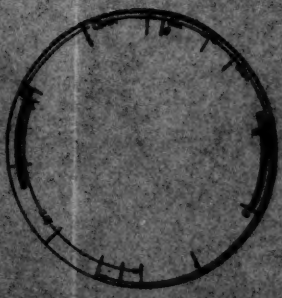












10.

in Figure 17, the spray valves are fueled in two groups, 6-2-8-5 and 9 in one group and 1-4-10-3 and 7 in another group. Hence, these simultaneous events have to be considered.

Figure 19 may be cut and mounted as indicated in Figures 17 and 18 for consideration of the following discussion, or, reference of statements to Figures 17 and 18 may be sufficient.

If the crank diagrams are set with sequence running in direction of rotation as in Figure 17, the *timing* sequence obtains and any radius on the chart shows what is occurring in all cylinders in terms of any cylinder. As an example, note that when Number 1 crank is at firing center:—

Number— 6 is in compression,
2 is in compression,
8 is in intake,
4 is in intake,
10 is in intake-exhaust,
5 is in exhaust,
9 is in exhaust,
3 is in exhaust, and
7 is in expansion.

From this it is clear that the simultaneous fuel pump delivery arcs can be laid down on the chart in their respective diagrams. It is necessary only to properly time the position of one plunger to its own working cylinder crank. The fuel pump timing is given for Number 1 cylinder in this case so it has been used. The heavy black segments indicate the fuel delivery arcs.

The pump, in this case, runs at crank shaft speed, hence delivers two charges per working cylinder cycle. The pump groups are 180 degrees apart; so for the 720 degrees engine cycle, four fueling arcs appear, five cylinders covering the same arc in each case. In addition to the simultaneous events occurring in all cylinders as shown above, the diagram now shows the simultaneous pump events.

If now the individual diagrams be moved so as to place all cranks at compression dead center for Number 1 cylinder, as in Figure 18, a picture obtains of the total engine timing, showing the time each spray valve receives its fuel in addition to the regu-

lar cyclic events in the working cylinder. The outstanding thing to be noted in this particular engine is that no spray valve receives fuel while injection is in progress.

THE RESIDUAL OIL DETERMINING CHART (see Figure 20).

As shown in the discussion of the engine cycle diagram (Figure 17), that diagram, when set with the cranks in timing sequence (opposite to regular firing sequence), shows on any radius the simultaneous events of the fuel system and the engine cycle. Therefore, if some system of hatching be used to show what has happened in the spray valve between fuel charges, any radius will show the exact status of fuel charges in all spray valves for that instant. If, then, Number 1 crank be assumed in any position, the radius through that point will pass through fueling areas in the various cylinders and indicate exactly the amount of fuel in each.

To consider this a little more closely, consider the circle for Number 1 cylinder. After injection there is a period of 46 degrees when the spray valve has no fuel in it. Then it receives during the next 90 degrees of crank angle its first charge of fuel for the cycle. During the succeeding 270 degrees this spray valve carries this first charge or really one-half of a total charge of fuel. Then, during the next 90 degrees of crank angle the spray valve receives its second charge of fuel. This is complete when the working cylinder crank is at bottom dead center with compression beginning. Through the succeeding 166 degrees of the cycle the valve carries a full charge of fuel. At this point (14 degrees ahead of firing center) injection commences. Therefore, there are six periods to be considered:

(1) The injection period.

(2) The period between the end of injection and the beginning of the first fuel charge arc.

(3) The first fuel charge arc.

(4) The period between the first and the second fuel charge arcs.

(5) The second fuel charge arc.

(6) The period between the end of the second fuel charge arc and the beginning of injection.

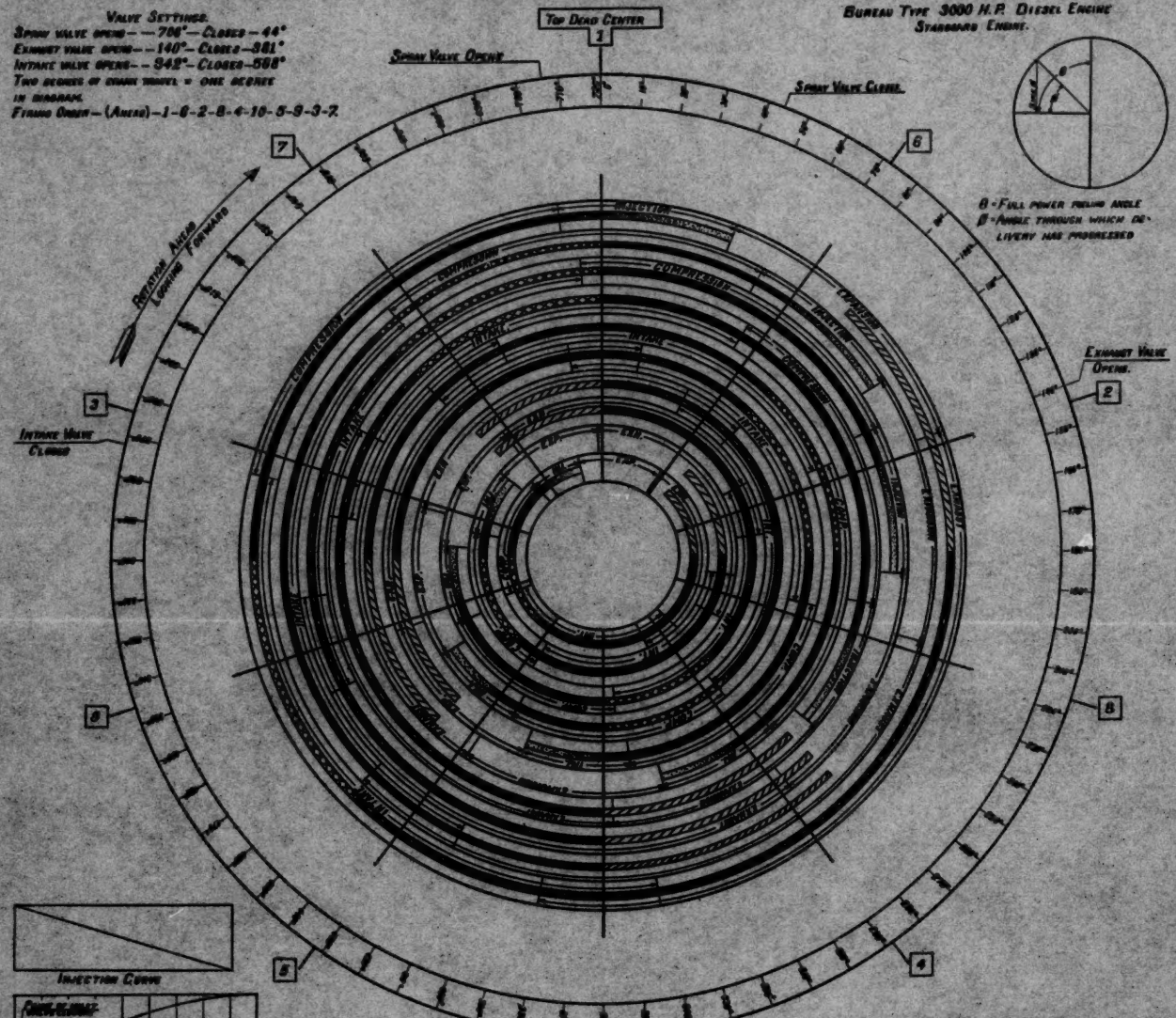
# RESIDUAL OIL DETERMINATION CHART.

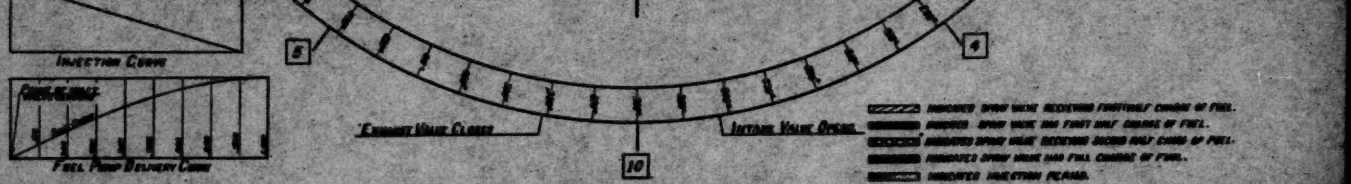
**Valve Settings.**
Spray valve opens — 786° — Closes — 44°
Exhaust valve opens — 140° — Closes — 381°
Intake valve opens — 342° — Closes — 568°
Two degrees of crank travel = one degree in diagram.
Firing Order — (Ahead) — 1-6-2-8-4-10-5-9-3-7

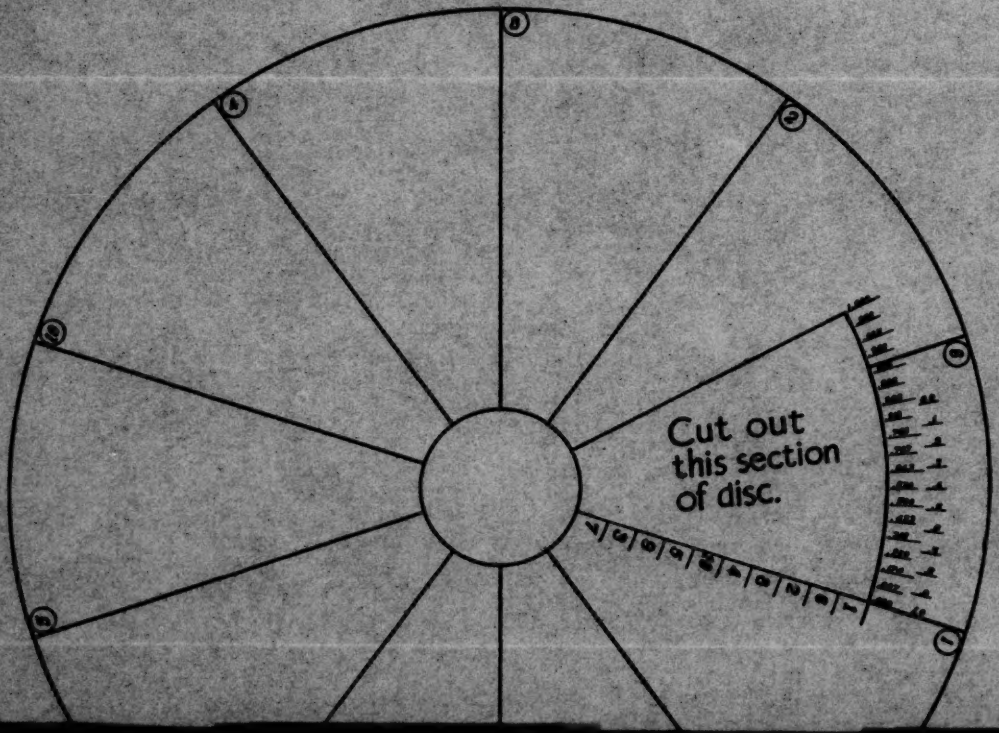
Bureau Type 3000 H.P. Diesel Engine
Starboard Engine.

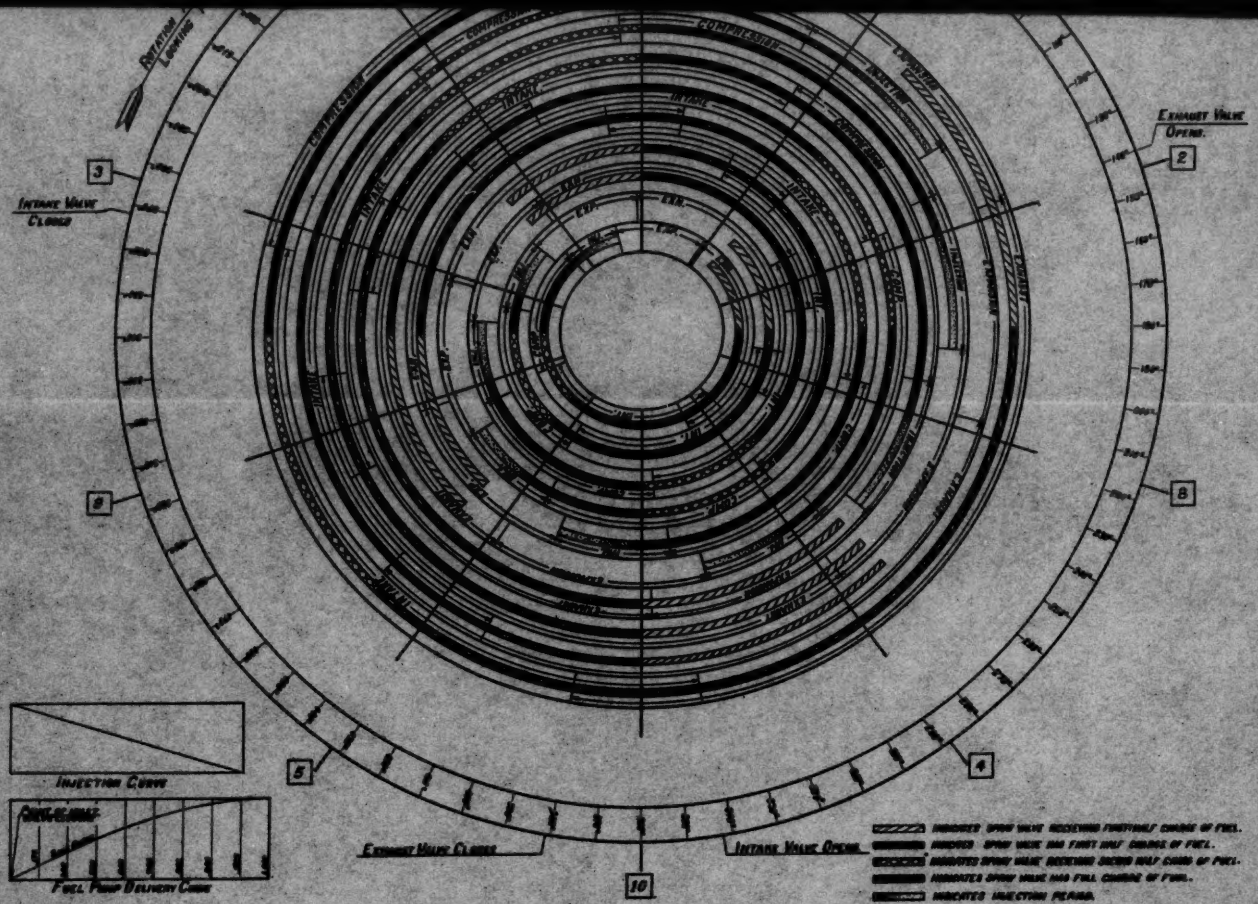


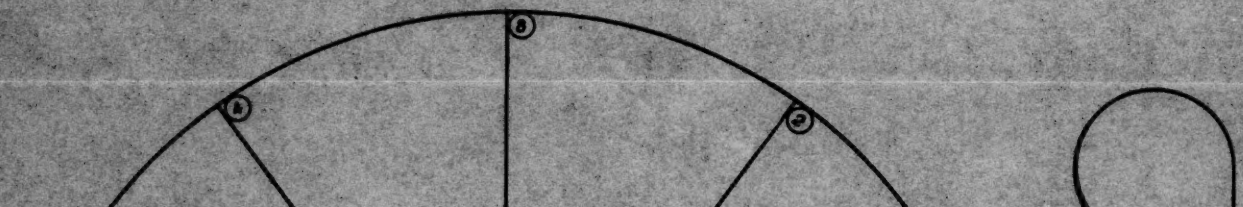

θ = Full power fueling angle
β = Angle through which delivery has progressed

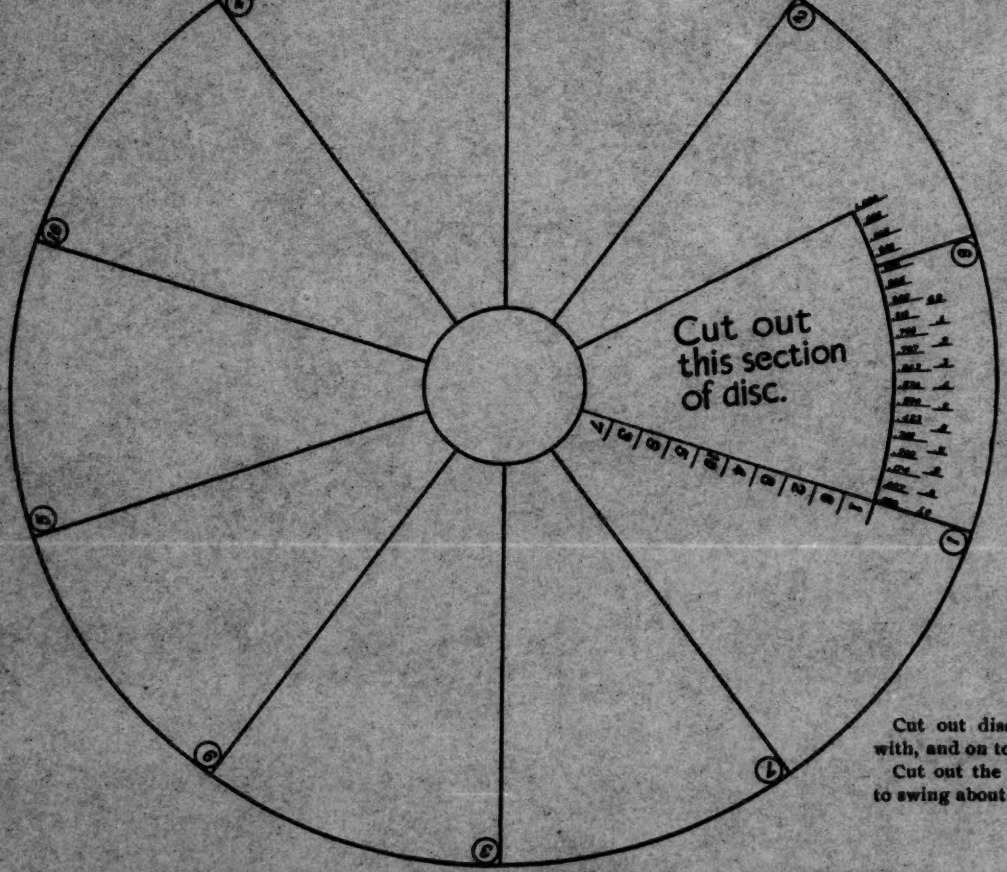


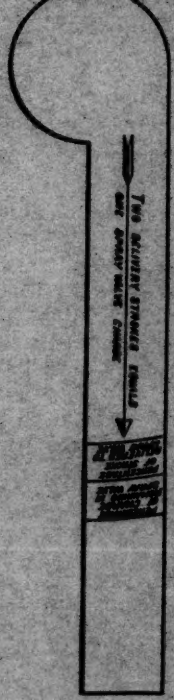

Injection Curve
Fuel Pump Delivery Curve
Exhaust Valve Closes
Intake Valve Opens
10


Cut out
this section
of disc.


Rotation Locking
Intake Valve Closed
Exhaust Valve Opens
Injection Curve
Fuel Pump Delivery Curve
Exhaust Valve Closes
Intake Valve Opens
Indicates spray valve receiving first half charge of fuel.
Indicates spray valve has first half charge of fuel.
Indicates spray valve receiving second half charge of fuel.
Indicates spray valve has full charge of fuel.
Indicates injection period.





Cut out disc and mount concentric with, and on top of the main chart.

Cut out the straight edge and mount to swing about center of disc.

FIGURE 20.

The hatchings for all of these periods except Number 2 are shown in the lower right corner of the chart of Figure 20, the space in the case of Number 2 being left blank.

The basic part of this chart is the same as for the engine cycle diagram of Figure 17 except that the fuel indication hatching has been changed. Having this hatching, by laying a straightedge down on a radius and noticing the points it passes through in the various fueling arcs, the amount of fuel in the various spray valves can be roughly determined. This operation is facilitated with a disc arranged to be put on top of, and concentric with, the main chart and movable around a common center. This disc shows only a small arc of the main chart. On this disc the radii representing cranks are laid out with firing sequence numbers running against rotation. Beginning with the radius for crank Number 1 and running backward against rotation through an arc length of 90 degrees of crank angle, a segment of the disc is removed. Along radius Number 1 of the disc, the firing sequence is laid out in same manner as on the engine cycle chart discs.

The arc corresponding to 90 degrees of crank angle is removed from the chart because the fuel pump delivers through this angle at full power. The radius corresponding to Number 1 crank is used as reference because it is convenient to refer events to positions of Number 1 crank.

Cut out the disc and straightedge from Figure 20 and mount as directed by note at bottom right-hand corner of the figure.

Suppose that Number 1 crank is at bottom dead center, i.e., has traveled 180 degrees from firing center. At this point the fuel pump has completed the delivery of one charge (See Figure 17) of fuel to spray valves 1, 4, 10, 3 and 7. Note that the segment which is removed from the disc just covers the fueling segment for this group of plungers. Consider, now, Number 1 crank at 160 degrees from firing center. At this point the fuel pump has 20 degrees to go to complete the charge to valves 1, 4, 10, 3 and 7. Here it will be noted that the segment removed from the disc runs back 20 degrees past the point where delivery to spray valves 1, 4, 10, 3 and 7 began. In this manner a glance shows the approximate state of completion of charge to the spray valves, for any crank position.

Where, as in this case, the fuel pump for full power uses the last quadrant of the discharge stroke for delivery, the percentage of charge in terms of delivery arc is represented by the sine curve. This curve is shown at the lower left corner of the chart, and the elementary pump diagram is shown at the upper right corner of the chart. If the 70-degree angle of the delivery arc mentioned above (i.e., Number 1 crank at 160 degrees) be applied to the sine curve, it is seen that the charge at that crank position is 94 per cent complete. Any angle can be measured and applied to the curve to determine the state of completion of the charge. This operation is shortened by laying down along the outer arc of the disc segment a sine table which runs backward from Number 1 crank to 90 degrees of crank angle. A straightedge, pivoted at the center of the chart and running out on the radius, is provided for taking readings. Placing this edge in contact with the point where the fuel arc begins, with Number 1 crank at 160 degrees as before, the edge strikes .940 of the sine table on the arc, which indicates 94 per cent of a charge. Take another example, supposing Number 1 crank to be 130 degrees past firing center. Setting the straightedge in contact with the beginning of the fuel arc, the edge cuts the sine table at .643, which is 64.3 per cent of a charge. Likewise any other percentages may be found.

The spray valve lift, in this case, is controlled, and for other reasons set forth in connection with Figure 12 the fuel delivered should vary approximately with the injection arc. Hence, fuel remaining in the spray valve during injection is indicated by a straight line. This is also shown at lower left of the chart. As in the case of the fuel pump delivery angle, the percentage of a charge in the spray valve for any crank angle within the injection angle is arranged along the outer arc of the disc segment.

The two scales, which are arranged along the outer arc of the disc segment, progress in opposite directions because the inner one represents oil being added to the spray valve, while the outer circle represents oil remaining in the spray valve after a part has been injected. The arc covered by the outer scale is equal to the arc of the injection angle.

There is an important difference in the two scales in the case of the four cycle engine using a fuel pump which runs at crank shaft

speed. In this case, which is the present one, there are two pump delivery strokes per engine cycle. Therefore, the percentages of pump delivery charges are based on one-half of the total spray valve charge. This difference does not occur in case the fuel pump is driven at engine cycle speed.

An example will now be worked with the chart to determine the amount of fuel in the whole engine for different positions of Number 1 crank. Assume Number 1 crank at 155 degrees from firing dead center and taking note of the hatching designations:

Spray valve No. 1 has received 90.6 per cent of its first pump charge, which is $\frac{90.6}{2}$ per cent of a full spray valve charge = 45.3 per cent.

No. 6 has injected its fuel and has received none = 00.0 per cent.

No. 2 is injecting fuel, but has left = 57.0 per cent.

No. 8 has full charge = 100.0 per cent.

No. 4 has one complete half charge plus 90.6 per cent of its last half charge = $\frac{100 + 90.6}{2}$ per cent = 95.3 per cent.

No. 10 has same as 4 = 95.3 per cent.

No. 5 has one complete half charge = 50.0 per cent.

No. 9 has same as 5 = 50.0 per cent.

No. 3 has same as 1 = 45.3 per cent.

No. 7 has same as 1 = 45.3 per cent.

Total amount of residual oil in engine with Number 1 crank at 155 degrees = 583.5 per cent, or $\frac{582.5}{10}$ per cent of a total engine charge.

The result, for similar determinations all around the chart, is shown in the dotted curve at the top of Figure 21.

It is to be noted that the above values are based on the assumption that the governor can stop delivery to a spray valve after the

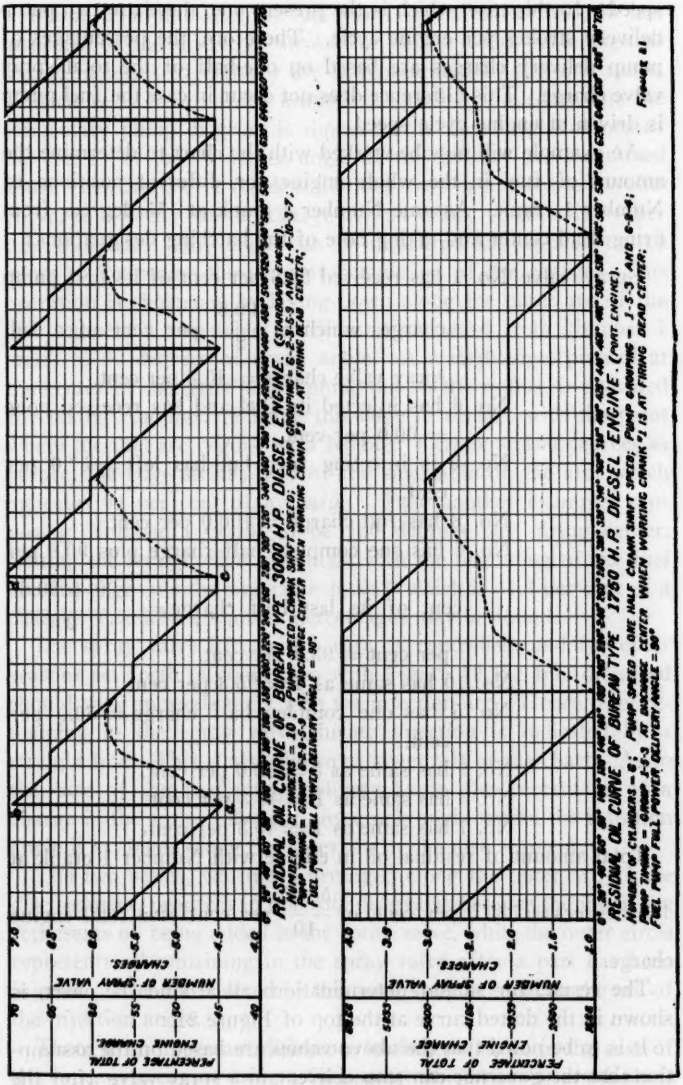

RESIDUAL OIL CURVE OF BUREAU TYPE 3000 H.P. DIESEL ENGINE. (STANDARD ENGINE).
NUMBER OF CYLINDERS = 10; PUMP SPEED = CRANK SHAFT SPEED; PUMP GROUPING = 6-2-1-5-3 AND 7-4-10-9-8;
PUMP TIMING = GROUP 1-5-3 AT DISCHARGE CENTER WHEN WORKING CRANK "1 IS AT FIRING DEAD CENTER;
FUEL PUMP FULL POWER DELIVERY ANGLE = 30°.
NUMBER OF SPRAY VALVE CHAMBER.
PERCENTAGE OF TOTAL ENGINE CHAMBER.
RESIDUAL OIL CURVE OF BUREAU TYPE 1760 H.P. DIESEL ENGINE. (FRONT ENGINE).
NUMBER OF CYLINDERS = 6; PUMP SPEED = ONE HALF CRANK SHAFT SPEED; PUMP GROUPING = 1-5-3 AND 6-2-4;
PUMP TIMING = GROUP 1-5-3 AT DISCHARGE CENTER WHEN WORKING CRANK "1 IS AT FIRING DEAD CENTER;
FUEL PUMP FULL POWER DELIVERY ANGLE = 30°.

FIGURE 81

pump has actually started its effective delivery stroke. Since this action cannot obtain in this engine, due to the reactive pump control valves used, the curve is shown dotted at sections where the assumed non-reactive type of pump control valves make a difference. The dotted curves serve only to accentuate the difference due to use of reactive valves.

The full lines of this curve, and of all other similar curves, indicate the actual residual oil resulting from the combinations shown. In determining values for this curve, no notice need be taken of the inner segment scale on the residual oil determining chart indicating the "Percentage of stroke delivered to spray valves," since with reactive type valves a charge started to spray valve must be completed. Therefore, at any position within a delivery arc, the total charge must be counted. This means that entire groups must be added at the instant of suction valve closure as shown in curve at *a-b, c-d,* etc. Hence, at point "*a*" five full pump charges (two and one-half spray valve charges) are added to lift curve to point "*b*." This curve shows that when running at full power the engine has a minimum of 45 per cent of a total charge, and a maximum of 70 per cent of a total charge. There are two maxima and two minima per cycle. Inspection of Figure 17 will show that residual oil can be slightly reduced by exchanging leads 1 and 8 or 8 and 10.

The spray valves used in this engine are of the type of Figure 15-A. As shown in the discussion in connection with Figure 15, the fueling angle may lap the total injection angle if the pump runs at one-half crank shaft speed, and if the valves used are of type "A" of that figure. But the pump for the Bureau type 3000 H.P. engine runs at crank shaft speed. It, therefore, delivers two charges per engine cycle. These charges are separated by time interval of 90 degrees. In view of this it is probable that at low powers the fuel distribution would be uneven if the fueling angle lapped the total injection angle. Suppose, for example, that the engine were running at 20 per cent of its full indicated H.P. Then the first charge (10 per cent) would run down in the valve and probably be blown into the cylinder as a pilot charge. This would probably take place during the first 15 degrees of injection angle. The fuel pump would start to deliver its last

charge about 25 degrees after the start of injection. There would then be 10 degrees of angle during which no fuel would be injected and during which blast chilling would take place. At medium and high powers this trouble would not exist, particularly with well regulated spray needle lift control.

For this engine the arrangement of Figure 15-3 should be satisfactory at all powers and reduce the residual oil maxima approximately 11 per cent.

For all of the following cases the residual oil values due to pump characteristics have been determined by charts similar to those of Figure 20.

The lower curve of Figure 21 is that for the Bureau type 1750 H.P. engine. The pump for this engine runs at half crank-shaft speed; so one charge of fuel is delivered to the spray valve per cycle. In this case a small injection lap exists in cylinders 1 and 6. All fueling arcs are moved as near injection as is possible without further injection lap. It appears advisable to increase the injection lap for cylinders 1 and 6 so as to cover at least one-half the injection angle as discussed under case 3 of Figure 15. In case close governing were paramount, total injection angle lap, as in case 1 of Figure 15, should be used.

### S-18 TYPE.

The curve for the S-1, S-4 to 9 and S-18 to 41 starboard engines is shown at the top of Figure 22. Residual oil could be reduced in this case in the same manner as for the Bureau 1750 type.

### R-1 TYPE.

The curve for the R-1 to 20 port engines is shown at the bottom of Figure 22. Figure 23 shows in "A" the original arrangement with its corresponding "A" curve. This arrangement appears to be particularly bad and is believed to be a drafting room mistake. In this case spray valve Number 1 receives fuel about 180 degrees after injection, whereas spray valve Number 4 has a fueling lap of 42.5 degrees. With the type of spray valve used, this would probably make little difference except in case of leaking spray valves. The arrangement of diagram "B" of this figure would provide greater fueling symmetry and the same

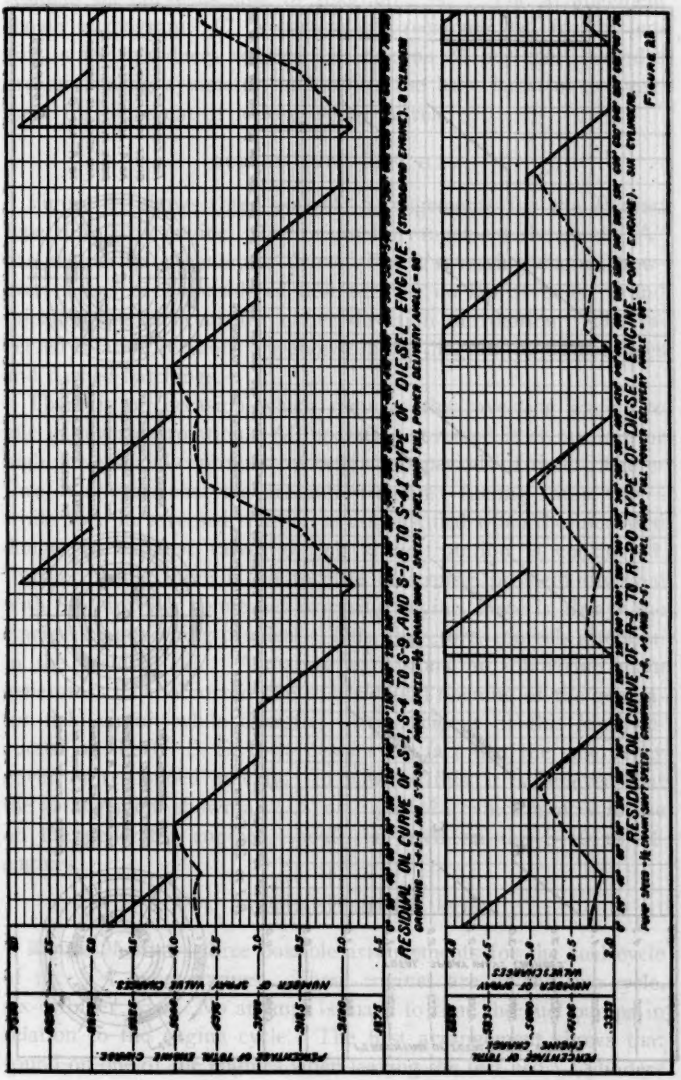

RESIDUAL OIL CURVE OF S-1, S-4 TO S-9, AND S-16 TO S-41 TYPE OF DIESEL ENGINE
RESIDUAL OIL CURVE OF R-1 TO R-20 TYPE OF DIESEL ENGINE

Figure 23

PERCENTAGE OF TOTAL CAM LIFT CHANGE.
TOTAL SPRAY VALVE CHANGE.
A
B
C
Driving
Coasting
A-1 TO 20 CAMS INDICATING POSSIBLE ARRANGEMENT IN RESIDUAL OIL CHARACTERISTICS DUE TO REGRINDING
THIS ARRANGEMENT GIVES RESIDUAL OIL CHARACTERISTICS SHOWN BY CAM "A". A FEELING LAP OF 43.5° OBTAINS. THIS = 21.5 PUMP INCHES = 7% OF CHANGE.
CHANGED GRINDING - REDUCES FEELING LAP GIVES SAME RESIDUAL OIL CHARACTERISTICS. THIS IS AN IMPROVEMENT SINCE IT PROVIDES GREATER FEELING SYMMETRY.
THIS CHANGE OF GRINDING, GIVES SAME FEELING LAP AS "A", BUT REDUCES RESIDUAL OIL VALVE EFFECTS, AND HENCE IS SUPERIOR TO "A" AND "B".
FIGURE 23

residual oil curve as that of diagram "A." The arrangement shown in diagram "C" is particularly recommended for this engine, since it reduces residual oil maxima 25 per cent, provides greater fueling symmetry, and offers the least harmful arrangement for the event of leaking spray valves.

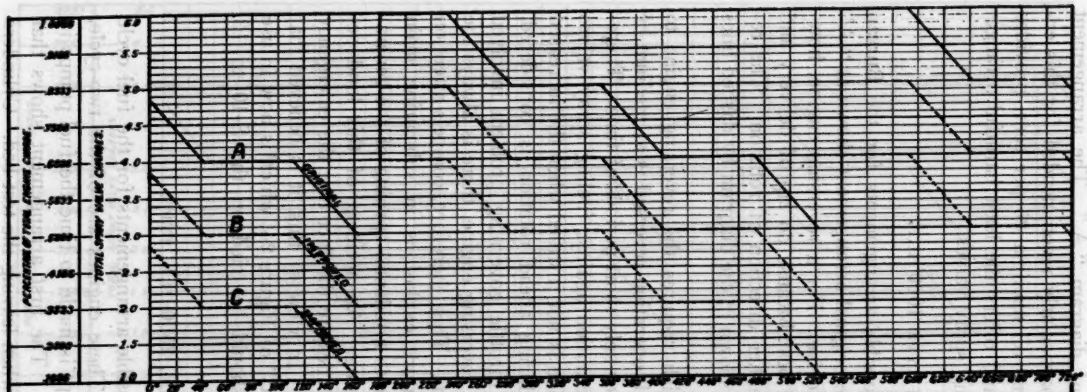
### BUREAU TYPE 1000 H.P.

Figure 24 shows three possible arrangements for the Bureau type 1000 H.P. engine. The original arrangement shown at "A" has no merit for generator drive. For a reversing engine, however, it has been considered best because no lap of delivery and injection angles occur for either direction of rotation. The reversed rotation relations are shown in the diagram by dashed lines and dashed hatchings.

There is no occasion which requires the reversing engine to idle in reversed rotation. All operation reversed is at medium or high powers, and continues for only short periods of time. Therefore, it appears that the precaution against fueling spray valves while open in reversed rotation is not justified when the increased danger to exhaust valves during ahead rotation is considered. Further, while there is no throttle governing problem involved with the propeller connected reversing engine, there is always the emergency governing problem. This, therefore, compels attention to the residual oil. Armature explosions in submarine main generators have occurred due to accidental opening of the generator field circuit while charging. An accident of this kind occurred on the U. S. S. R-25 despite the fact that the emergency governor responded quickly. In view of this, it is believed that the arrangement shown at "C" of Figure 24, which is now in use on the generator engines, should be tried on the S-10 to 13 engines.

### V-1, 2-CYCLE, 2500 H.P. TYPE.

Figure 25 shows three possible arrangements for the fuel cycle of the V-1 main engines. These engines are of the two-cycle, six-cylinder type. No attempt is made to time the fuel pumps in relation to the engine cycle. The first arrangement shows that found on one of the engines when leaving the test bed. Cylinders




CURVES OF BURMEISTER TYPE 1000 H.P. DIESEL ENGINE SHOWING POSSIBLE IMPROVEMENT IN RESIDUAL OIL CHARACTERISTICS DUE TO FLEXIGROUPING.

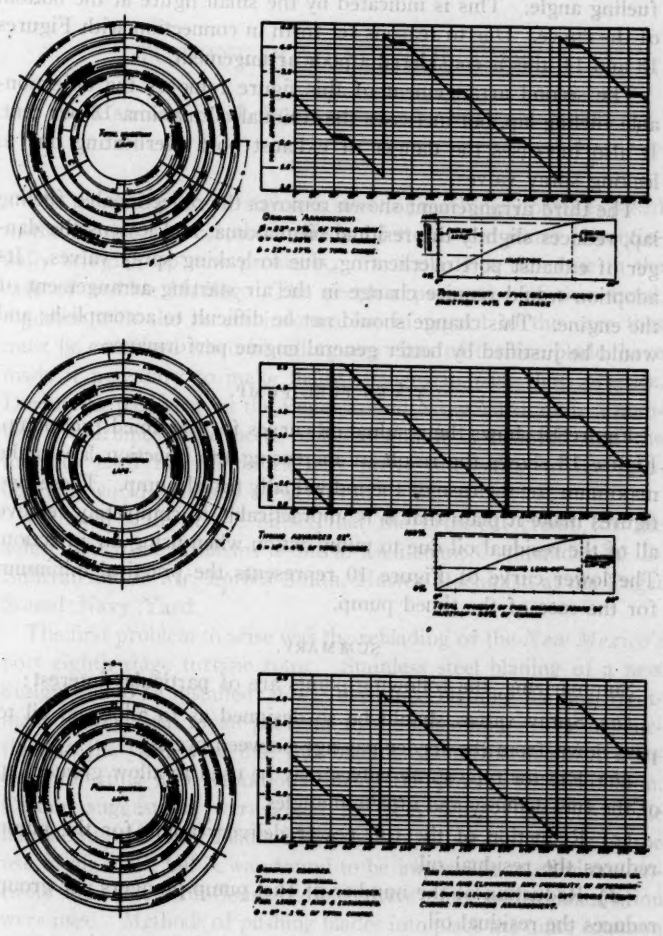
ORIGINAL ARRANGEMENT FOR REVERSING ENGINE.

THIS ARRANGEMENT GIVES RESIDUAL OIL CHARACTERISTICS SHOWN BY CURVE "A". EARLY FUELING WOULD CAUSE BURNING OF EXHAUST VALVES IN CASE OF LEAKING SPRAY VALVES.

THIS CHANGE OF GROUPING REDUCES RESIDUAL OIL MAXIMUM. DANGER OF BURNING EXHAUST VALVES IS CORRESPONDINGLY REDUCED.

THIS CHANGE OF GROUPING REDUCES RESIDUAL OIL MAXIMUM 33%. THE 8° FUELING LAP DOES NOT AFFECT THE RESIDUAL OIL MAXIMUM BUT DECREASES DANGER TO EXHAUST VALVES. THIS ARRANGEMENT IS SUPERIOR TO "A" OR "B".

FIGURE 24.



V-1 TO V-3 2500 B.H.P., 2 CYCLE, 6 CYLINDER ENGINE TIMING CHARTS WITH CORRESPONDING RESIDUAL OIL CURVES.

FIGURE 25

Numbers 2 and 5 have the total injection angle lapped by the fueling angle. This is indicated by the small figure at the bottom of the curve. Due to reasons set forth in connection with Figures 13 and 14, this is considered a poor arrangement.

The second arrangement of this figure removes the objectionable fueling lap, but increases the residual oil maxima 13 per cent. It also increases the danger of exhaust port overheating due to leaking spray valves.

The third arrangement shown removes the objectionable fueling lap, reduces slightly the residual oil maxima, and lessens the danger of exhaust port overheating, due to leaking spray valves. Its adoption would require change in the air starting arrangement of the engine. This change should not be difficult to accomplish; and would be justified by better general engine performance.

## TIMED FUEL PUMP.

Figure 10 shows the residual oil curves for the timed fuel pump. Figure 16 shows the result of increasing the injection lap to the maximum for the case of the individually timed pump. These two figures make it plain that it is impracticable to completely remove all of the residual oil due to pump action when using air injection. The lower curve of Figure 10 represents the practical minimum for the case of the timed pump.

## SUMMARY.

Summarizing, the following points are of particular interest:

(*a*) Spray valves should be so designed as to allow no oil to pass down from the feeder passage between cycles.

(*b*) Ejector type spray valves may be used to allow greater lap of the fuel delivery and injection angles.

(*c*) Reduction of the full power delivery angle for the pump reduces the residual oil.

(*d*) Reduction of the number of fuel pump plungers per group reduces the residual oil.

(*e*) The fuel pump timing should be retarded as much as practicable to reduce the residual oil to the minimum.

(*f*) The use of non-reactive type pump control valves is advisable where the governing demands are critical, if the character of the service permits.

# TURBINE BLADING PROBLEMS.

By Captain S. M. Robinson, U. S. Navy, Member.

The increasing use of stainless steel for machinery parts exposed to the erosive and corrosive effects of steam and water is continually bringing up new problems. Particularly is this true in the case of turbine blading. The peculiar properties of this metal, together with the recent information in regard to the care that must be exercised when installing any class of turbine blade, have made it necessary to make many changes in past shop practice. During the past year, at the Puget Sound Navy Yard, quite a number of turbines have been rebladed and it is believed that the methods used for this work may prove interesting to the shipbuilding world generally as well as to the naval service.

I should state at the outset that the principal credit for the solution of these problems is due to Lieutenant Commander W. D. Sullivan and to Mr. Forbes Smith, Master Machinist, at the Puget Sound Navy Yard.

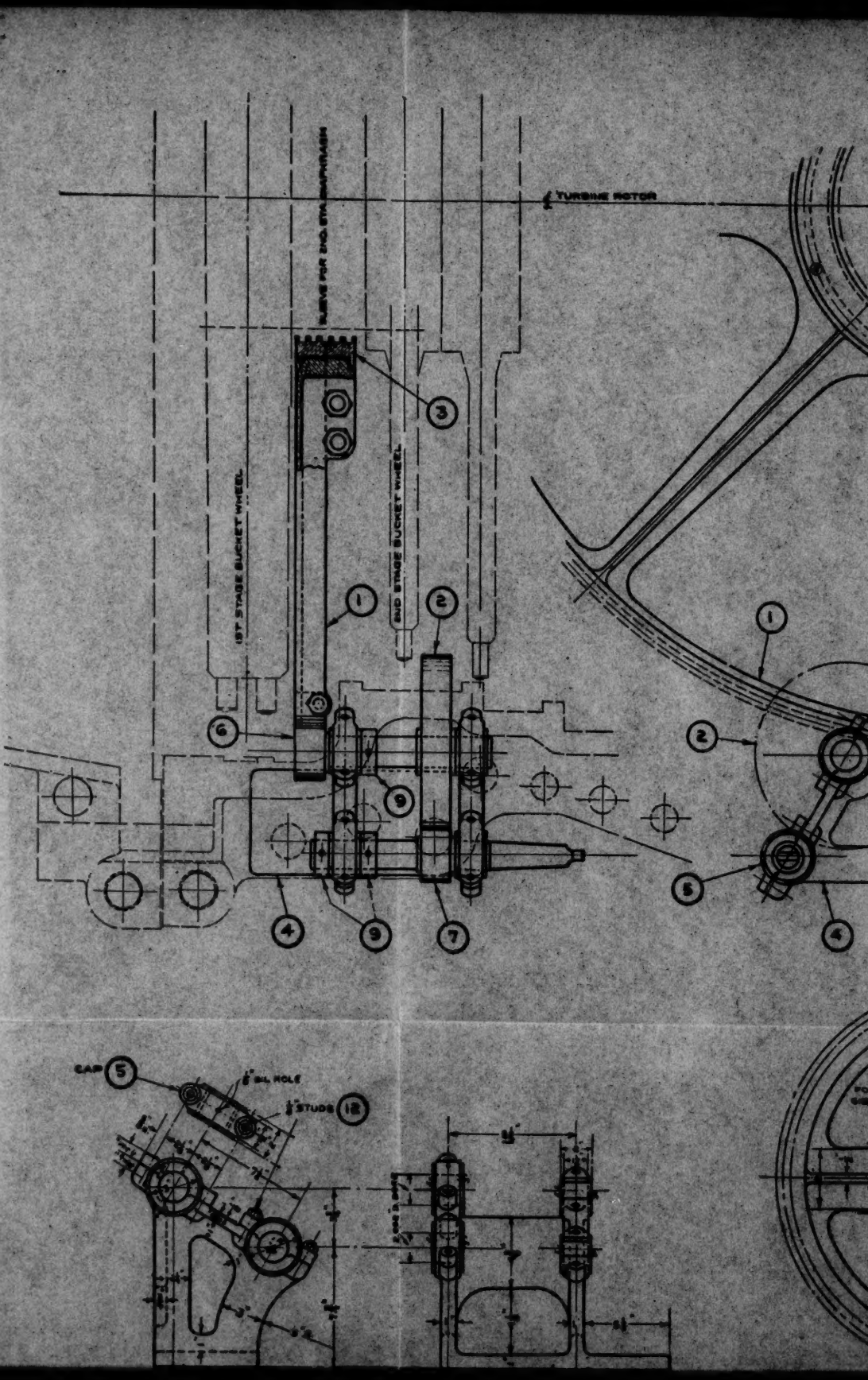
The first problem to arise was the reblading of the *New Mexico's* port eighth stage turbine rotor. Stainless steel blading of a new design was to be installed; it was machined blading, the heat treatment having been given previous to the machining. The instructions to the yard were that this blading should be installed without subjecting it to vibrations during the process of installation. Certain suggestions were given for preventing vibrations while hammering the rotor blades into position, either by hand or by the use of air tools, but it was found to be impossible to eliminate entirely all blade vibrations, if the "drive" method of installation were used. Methods of pushing blades into place are quite common in shop installations, but no portable device for doing this work on board ship had been perfected when this repair was undertaken, and in order to complete this installation in accordance with the Bureau's desire of eliminating blade vibration, a portable power

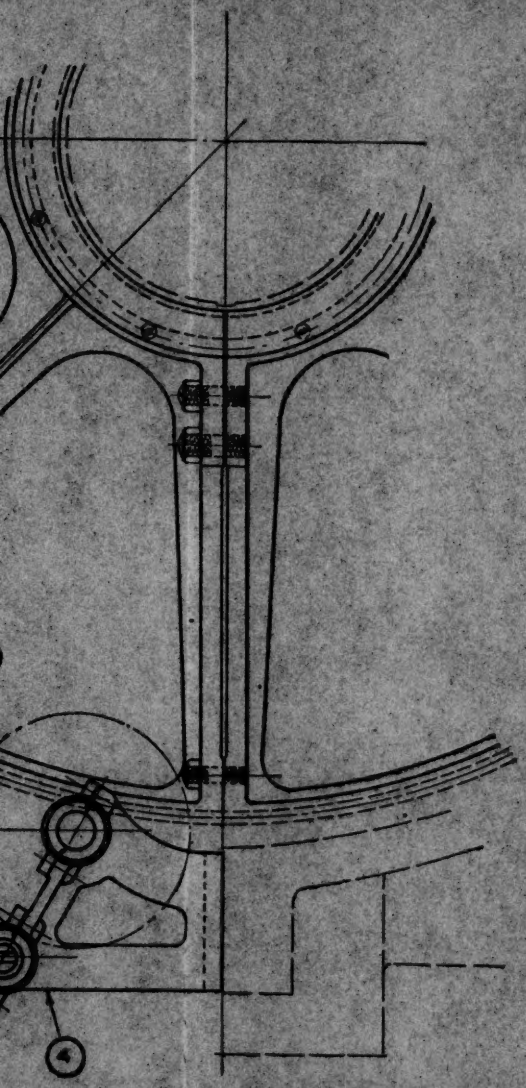
driver, as shown in Figure 1, was developed by this yard. It consists of a small double-reduction gear, the first reduction shaft being fitted for attaching an air motor and the second reduction gear being made in halves so that it can be attached to the turbine shaft or face of stage wheel; as shown in Figure 1, it is secured to rotor shaft, but where the design does not so permit, the split-bushings can be removed and the gear secured to stage wheel. As shown in Figure 1, this gear has a sleeve also in halves which fits over the diaphragm packing, and the gear wheel is then bolted over the sleeve, making a unit assembly; the bolts are adjusted so that this gear wheel will slip on its sleeve as described later. This wheel, as installed in Figure 1, replaces the diaphragm which has been rolled out.

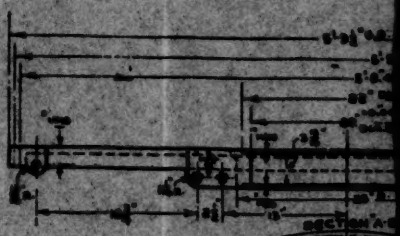
To use this blading machine, a blade is inserted in the circumferential blade groove of bucket wheel, and a driving tool held in a tool post and secured on the horizontal joint face of turbine casing is inserted behind it. The machine is then revolved, so that the blade will be pushed around in its groove till it comes up against another blade or a stop, at which time the large gear wheel will slip on its sleeve. The motor is then reversed and the rotor revolved back to position for inserting another blade; this absolutely eliminates vibration and insures uniform driving up of the blades. The adjustment of the friction between the gear wheel and its sleeve can, of course, be varied, and can be tested, if desired. This one size of machine can be used for reblading many different sizes of turbine rotors without other change than new sleeves for the gear wheel; by securing the gear wheel to stage wheel, or, by changing the gear wheel, practically any size of turbine rotor can be bladed.

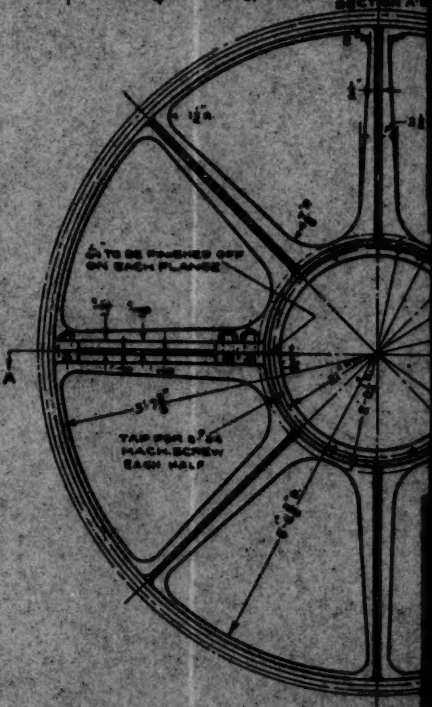
The next problem to arise was in connection with the shrouding for impulse blading. This problem is divided into two cases, one being shrouding secured by a tenon on the blade, and the other being shrouding secured in a blade groove.

The tenon type of blading will be considered first. Figure 2 shows a redesigned blade for the 16th stage of the *Farquhar*. The shrouding for this blading was also redesigned and of an experimental nature so that the yard was given complete instructions as to the method of installing it. Briefly, these were to countersink

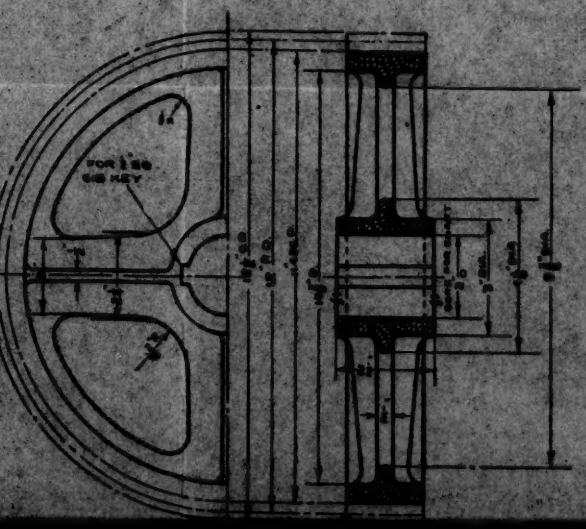

TURBINE ROTOR
SLEEVE FOR 2ND STAGE DIAPHRAGM
1ST STAGE BUCKET WHEEL
2ND STAGE BUCKET WHEEL
CAP 5
STUDS 12

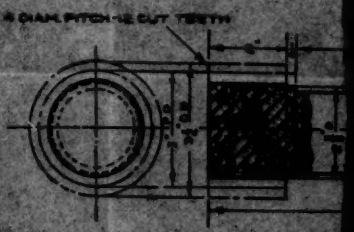


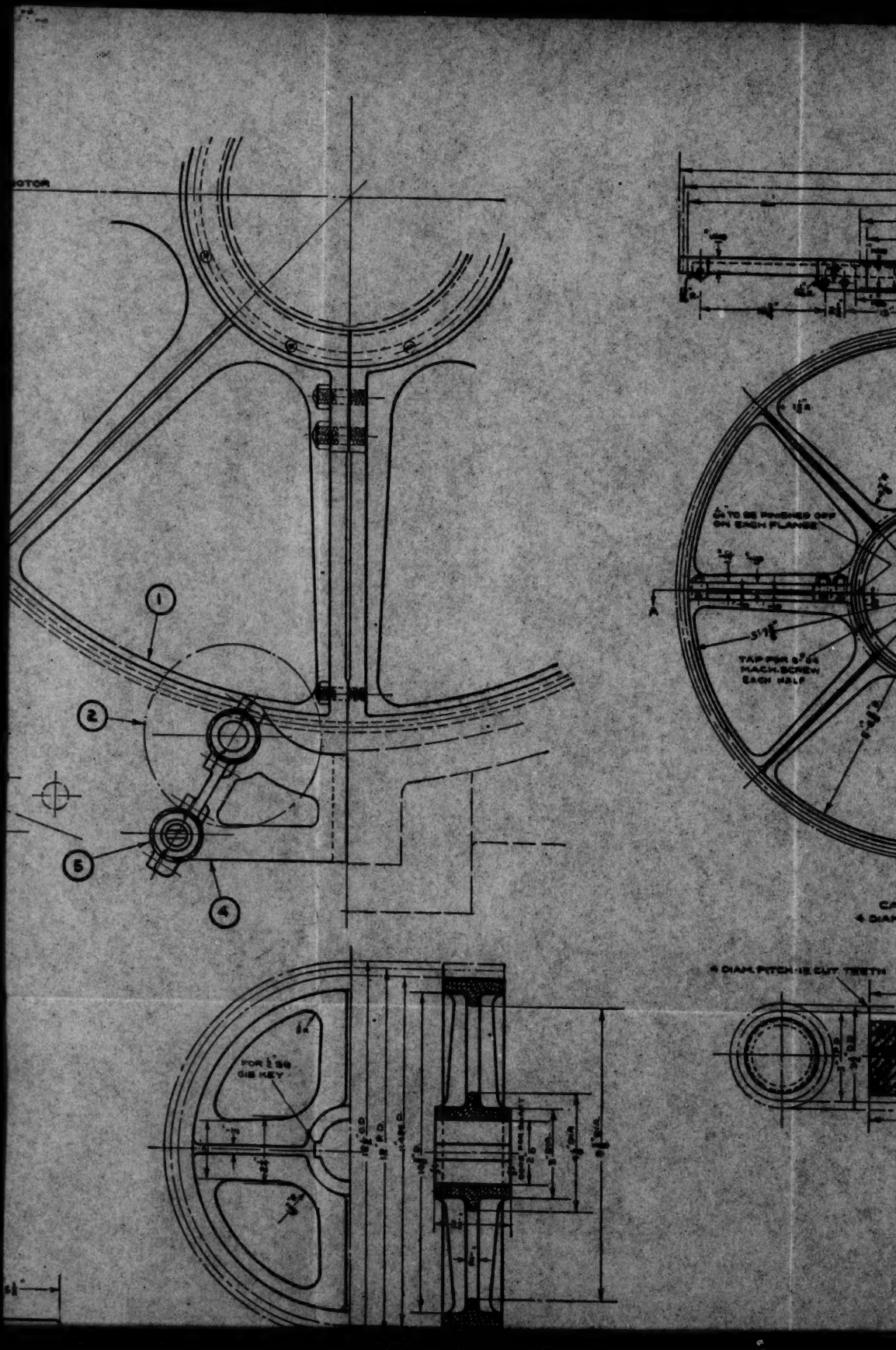


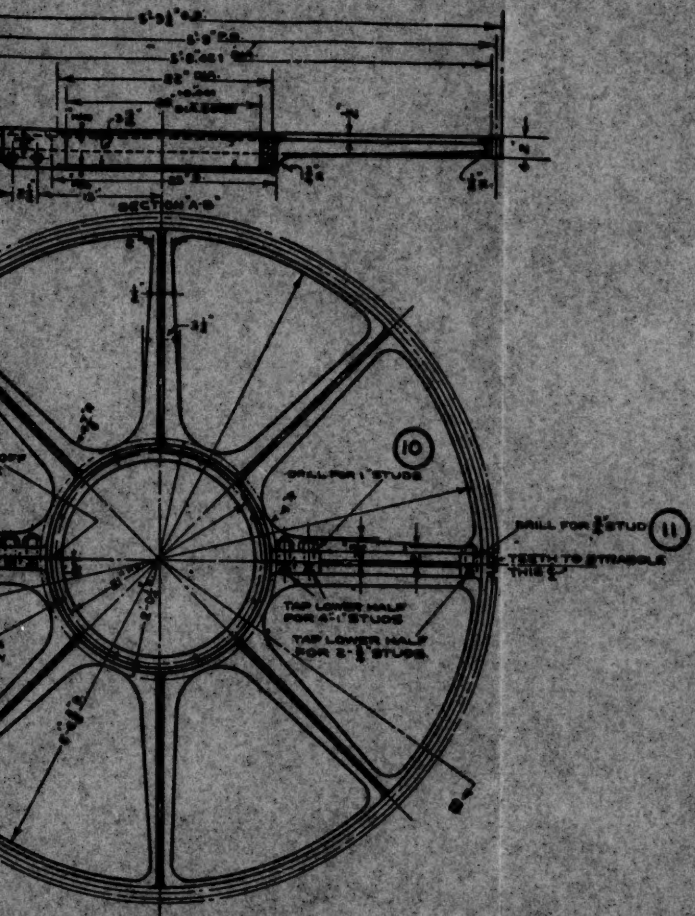



SPUR GEA
CAST IN HALV
4 DIAM. PITCH 27

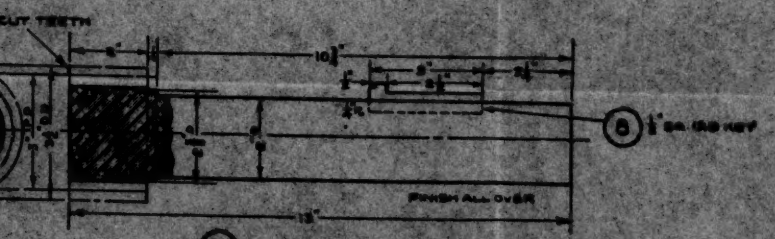


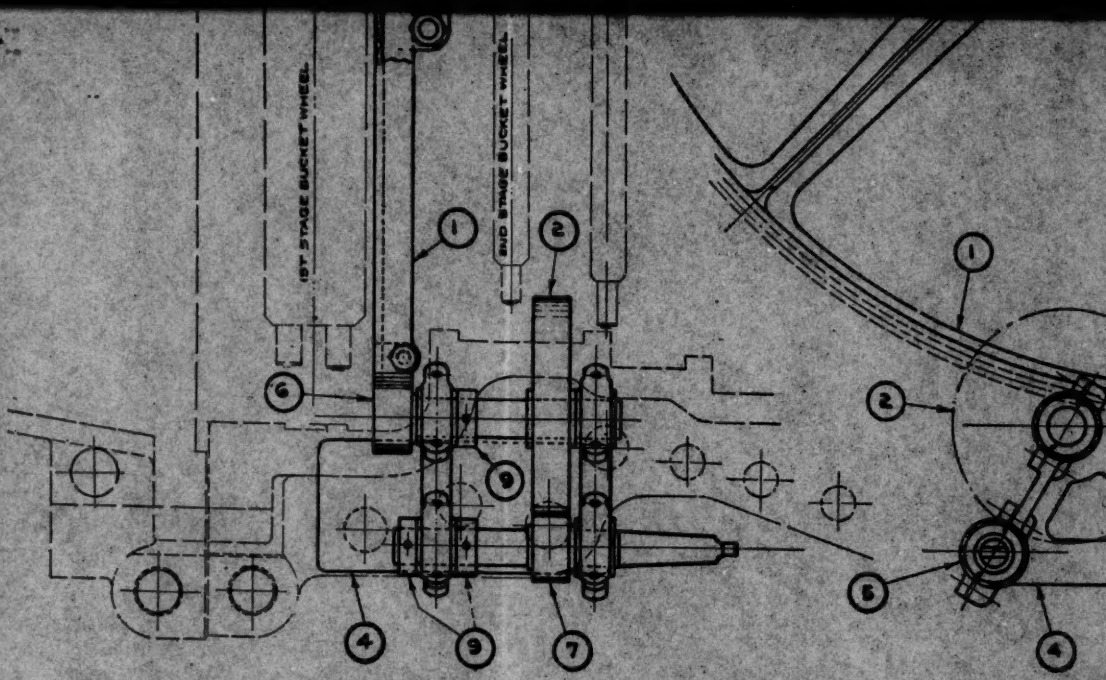

1
2
5
4
TAP FOR 0"25 MACH. SCREW EACH HALF
FOR 1"38 GIB KEY
4 DIAM. PITCH - 12 CUT TEETH

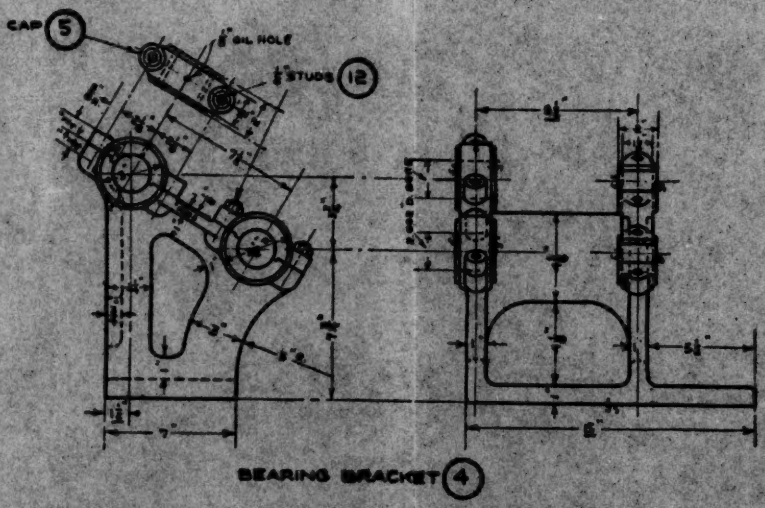


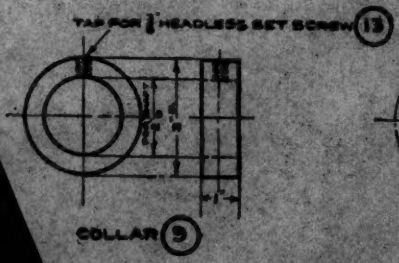

SPUR GEAR (1)
CAST IN HALVES
4 DIAM. PITCH .275 CUT TEETH

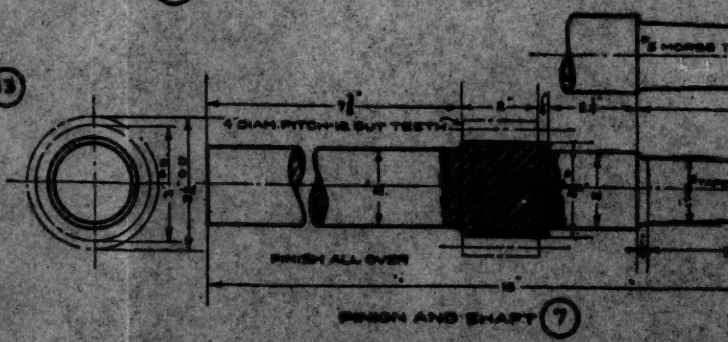


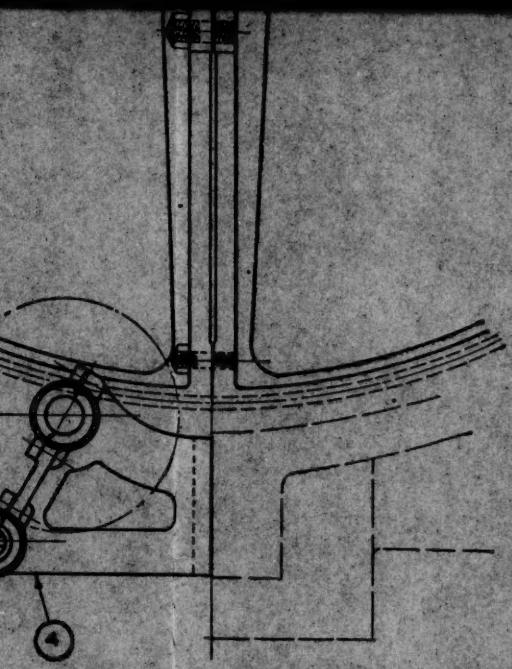

(6) PINION AND SHAFT

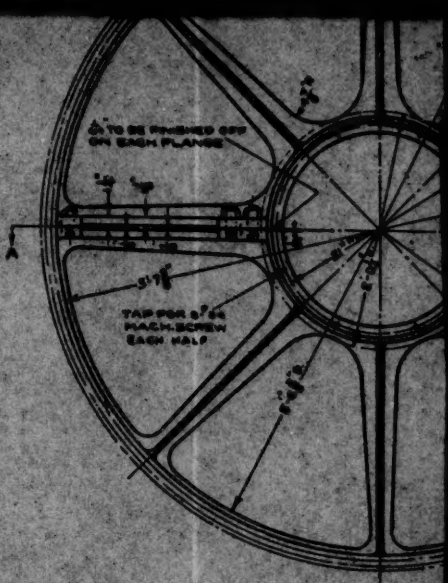

1ST. STAGE BUCKET WHEEL
2ND STAGE BUCKET WHEEL
1
2
6
9
4
3
7
1
2
5
4


CAP 5
OIL HOLE
STUDS 12
BEARING BRACKET 4

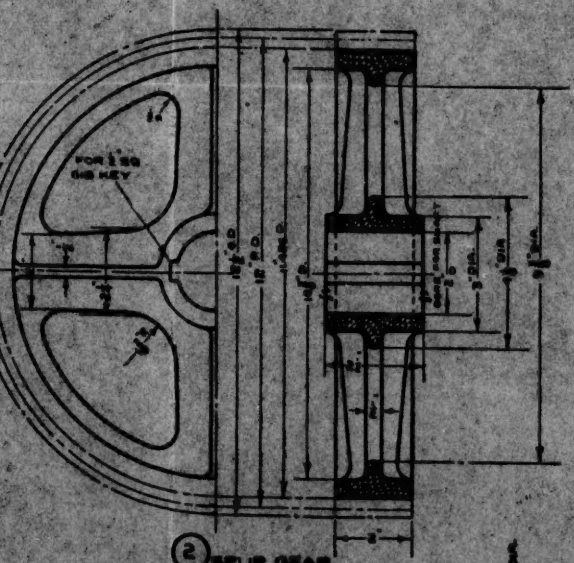

TAP FOR HEADLESS SET SCREW 13
COLLAR 9


DIAM. PITCH 18 CUT TEETH
FINISH ALL OVER
PINION AND SHAFT 7

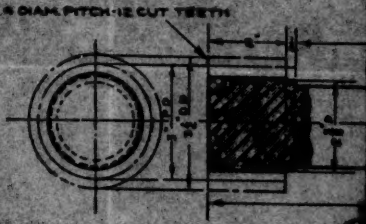


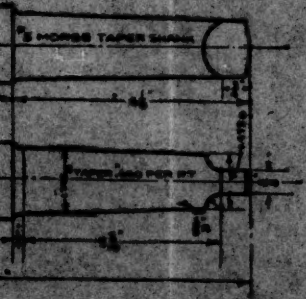


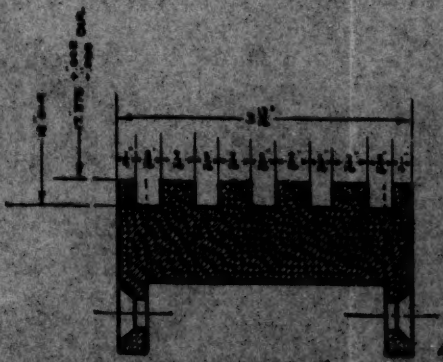

SPUR GEAR
CAST IN HALVES
4 DIAM. PITCH .275 C



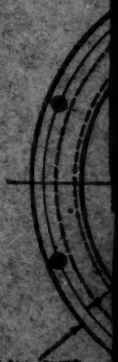

2 SPUR GEAR
4 DIAM. PITCH 48 CUT TEETH



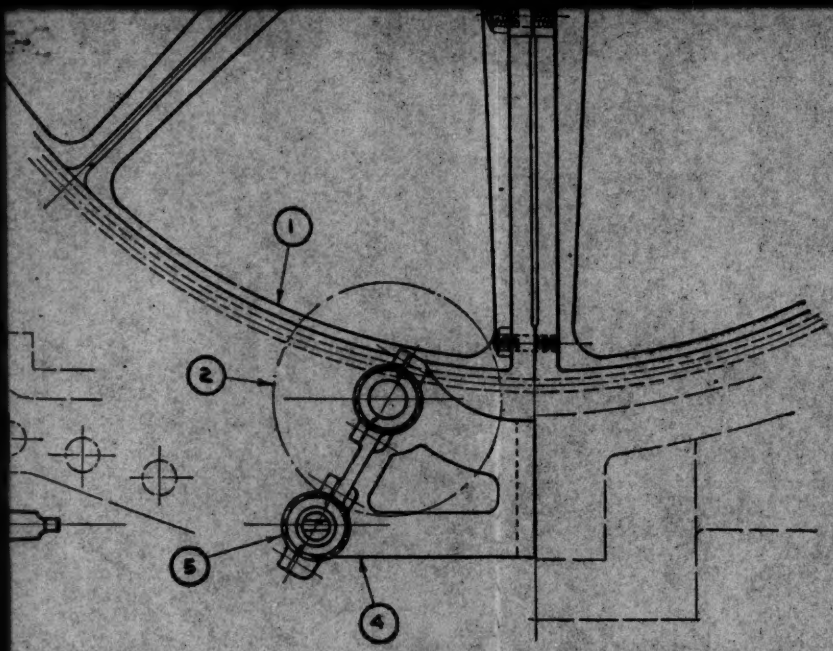



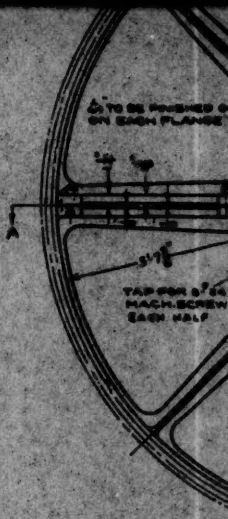
SECTION A-B
FULL SIZE

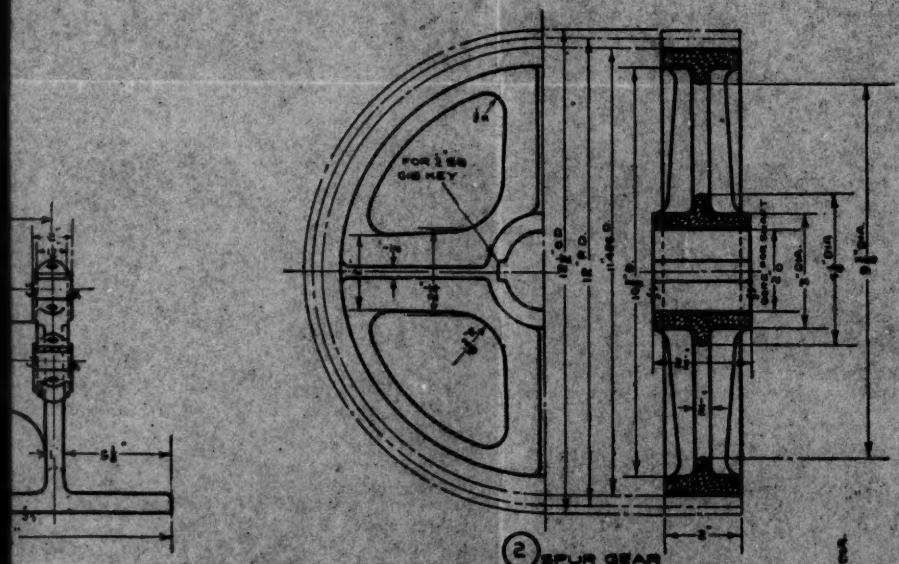



CAST IN ONE

AS DRAWN FOR U
ADAPTER SIMILAR

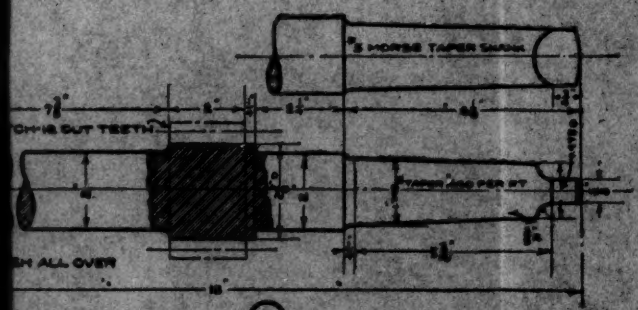


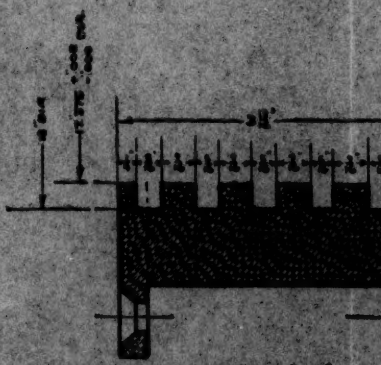



2 SPUR GEAR
4 DIAM. PITCH 48 CUT TEETH

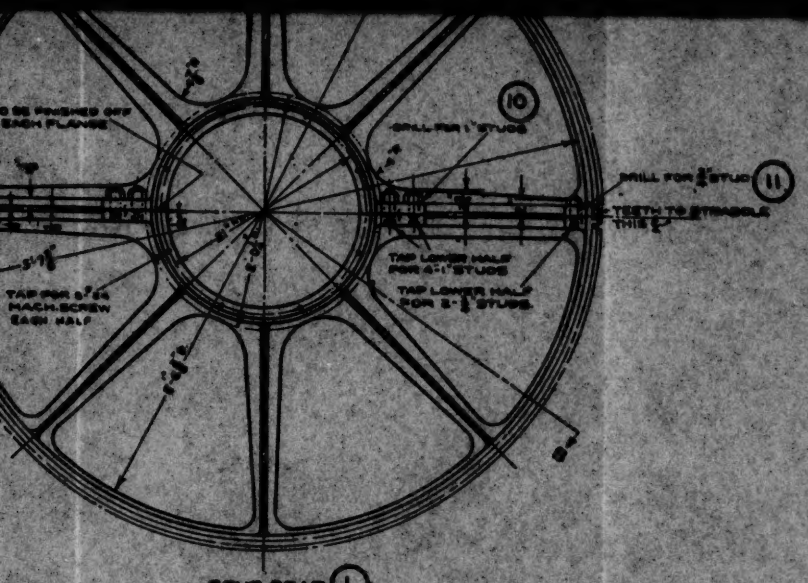




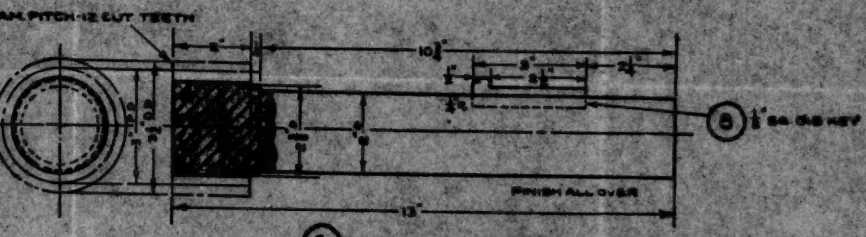

PINION AND SHAFT 7



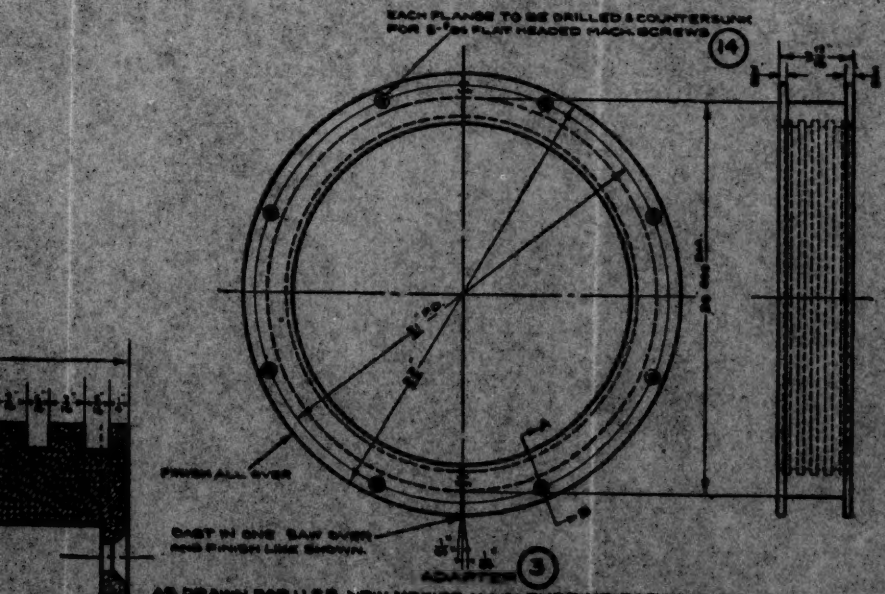
SECTION A-B
SCALE FULL SIZE




SPUR GEAR (1)
CAST IN HALVES
4 DIAM. PITCH .275 CUT TEETH

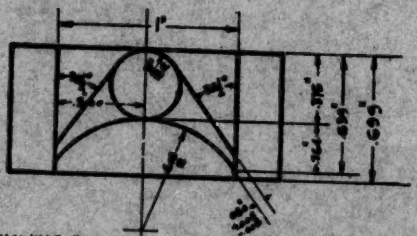



(6) PINION AND SHAFT


EACH FLANGE TO BE DRILLED & COUNTERSUNK FOR 8-1" FLAT HEADED MACH. SCREWS
14
FINISH ALL OVER
CAST IN ONE SAW OVER AND FINISH LIKE SHOWN

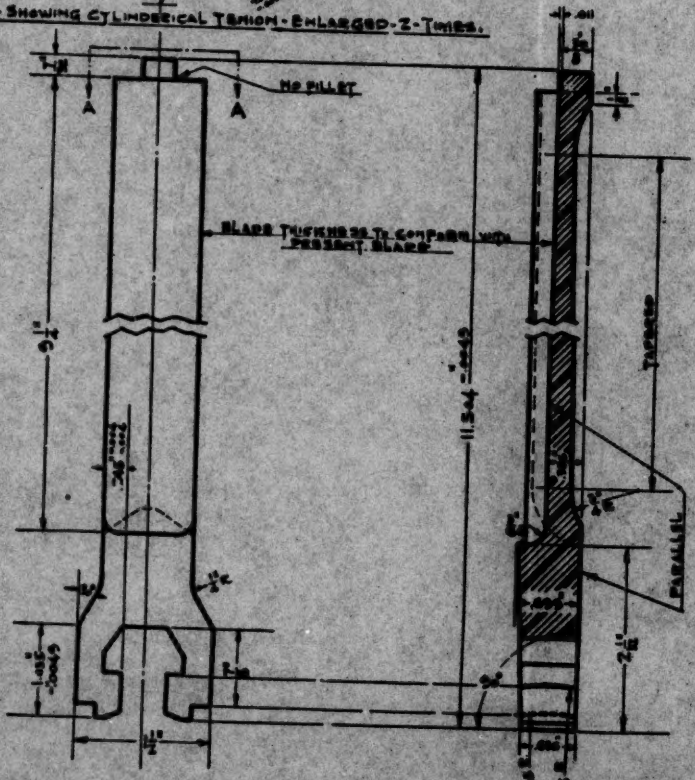
(3) ADAPTER

AS DRAWN FOR U.S.S. NEW MEXICO MAIN TURBINE ROTOR SLEEVE
ADAPTER SIMILAR FOR U.S.S. CALIFORNIA, MARYLAND AND WEST VIRGINIA

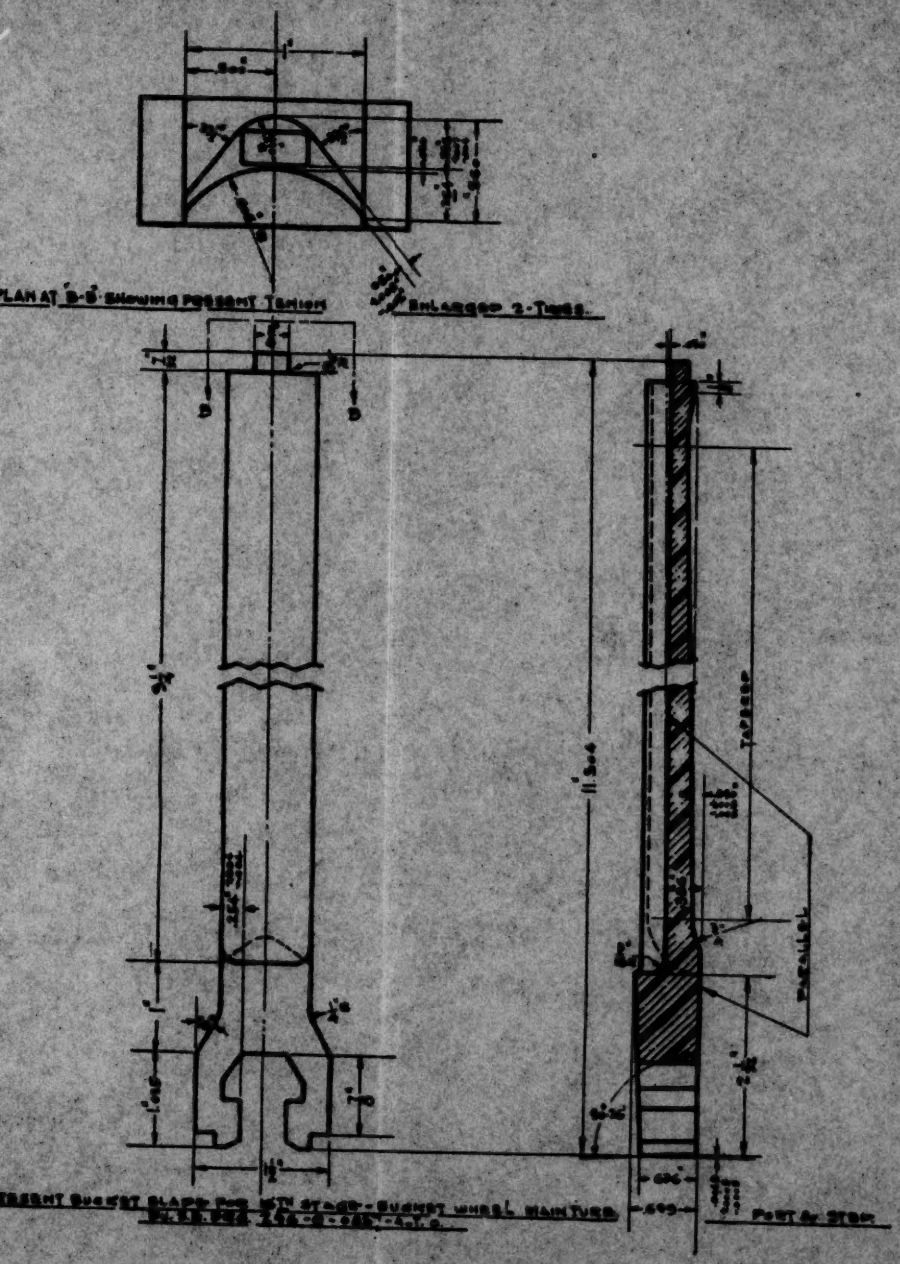



PLAN AT 'A-A' SHOWING CYLINDERICAL TENION - ENLARGED 2 TIMES.

PLAN AT 'B-B'




PROPOSED BUCKET BLADE FOR 10TH STAGE - BUCKET WHEEL, MAIN TURBINE PORT & STBD., PRESENT DU


PLAN AT B-B SHOWING PRESENT TENSION
ENLARGED 2-TUBES
D
D
TAPERED
PARALLEL
PRESENT BUCKET BLADES
BUCKET WHEEL MAINTYRE
POST & STOP

the rectangular holes in the shrouding on the top or outside and to fit the shrouds to the buckets exactly as the latter were assembled. Inasmuch as the redesign blades had fillets at the tenon root, it was necessary to countersink the holes on both sides.

It is believed that punching stainless steel will usually result in forming cracks in the corners of the holes and these cracks are very difficult to discover, as they are usually very fine. But, if cracks are formed, it is very probable that the shrouding will give trouble. It will be noted that the blade in Figure 2 has an enlarged tip section. This makes it possible to use a round tenon instead of one of rectangular section without reduction of cross sectional area. The holes in the shrouding can then be drilled and countersunk without danger of forming cracks. In addition, there are the following advantages in machining operations:

(*a*) *Tenon forming on turbine blades.*

In machining rectangular tenons, two machine operations are required; one to mill the tenon sides laterally, and one to mill them axially. Round tenons are formed in one operation with a hollow mill. The time and cost factor for this operation are in favor of the round tenon.

(*b*) *Punching shrouding for rectangular tenons.*

Due to the variations in rectangular tenon measurements, as found on different types of turbine blades, it is necessary to manufacture punches and dies for each tenon size. This cost would be eliminated by substituting round for rectangular tenons, as drills of any desirable diameters are always available. It is also possible to obtain a closer tenon fit with a round than with a rectangular tenon.

(*c*) *Countersinking shrouding.*

Forming countersinks for rectangular tenons resolves itself into hand operations, which are expensive, slow and unsatisfactory from the standpoint of finish and uniformity. Countersinking for round tenons would be performed with a machine countersink, assuring a true surface with a considerable saving in labor and time.

(*d*) *Riveting blade tenons.*

Riveting rectangular blade tenons is considerably more difficult than riveting round tenons for the following reasons:

Round tenons will fit the drilled shrouding better than rectangular tenons fit the punched shrouding; this is mainly due to the fact that quite often the blade tenon does not line axially with the axis of the rotor shaft, and unless the mechanic engaged in fitting the shrouding to the rectangular tenons is exceptionally careful, there will be an unequal clearance between the sides and edges of the tenon and shrouding. If the shrouding is forced on to avoid this condition, a strain is placed on the turbine blade and tenon by so doing.

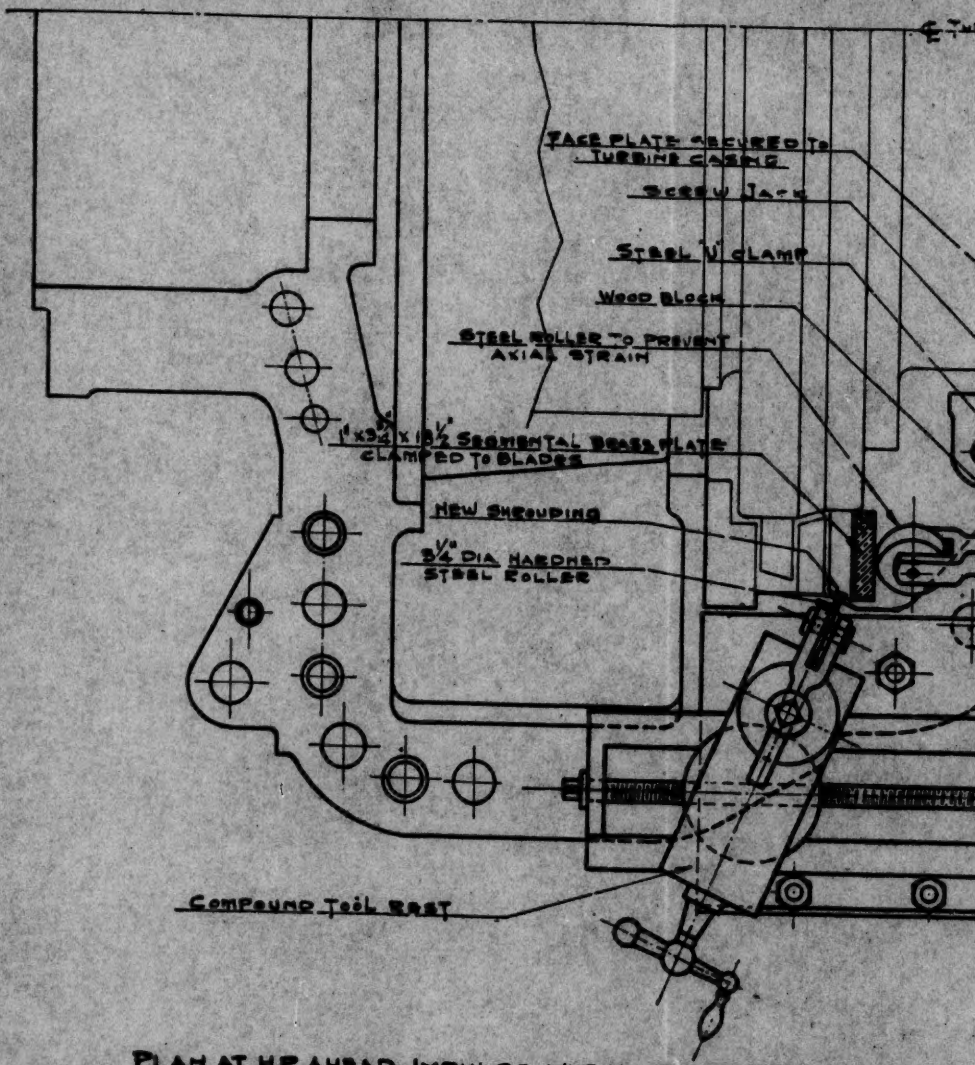
Round tenons and drilled shrouding will fit irrespective of blade position with the assurance of a 100 per cent circumferential tenon contact.

The rivets on rectangular blade tenons are formed with a flat faced set and a hand hammer; unless the mechanic exercises the utmost care, considerable damage is done to the face of the shrouding due to the edges of the hardened caulking tool coming in contact with the face of shrouding. The condition not only damages the shrouding, but is unsightly from the standpoint of finished work.

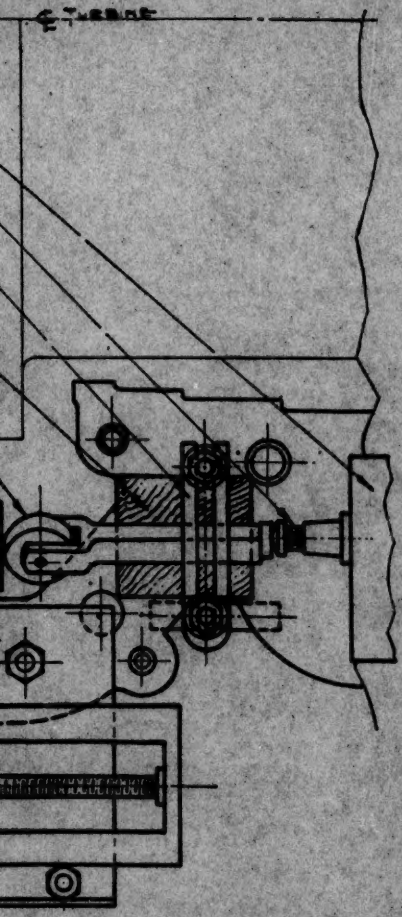
Riveting round tenons would be accomplished with a countersink caulking tool, which is easily operated and is held in position by the cone of the blade tenon. This would eliminate defacing the shrouding surface when riveted.

The difficulties involved in securing shrouding in grooves are much greater than in the case of the tenon type shrouding. This yard completed a repair job on Number 2 Fifth stage H.P. turbine rotor, U. S. S. *Omaha,* with this type of shrouding and the method will be described in detail. The original blading and shrouding were of nickel steel and the shrouding was caulked into grooves and secured by silver solder. Inasmuch as the new blades and shrouding were of stainless steel which has been heat treated, the silver solder method of securing was eliminated.

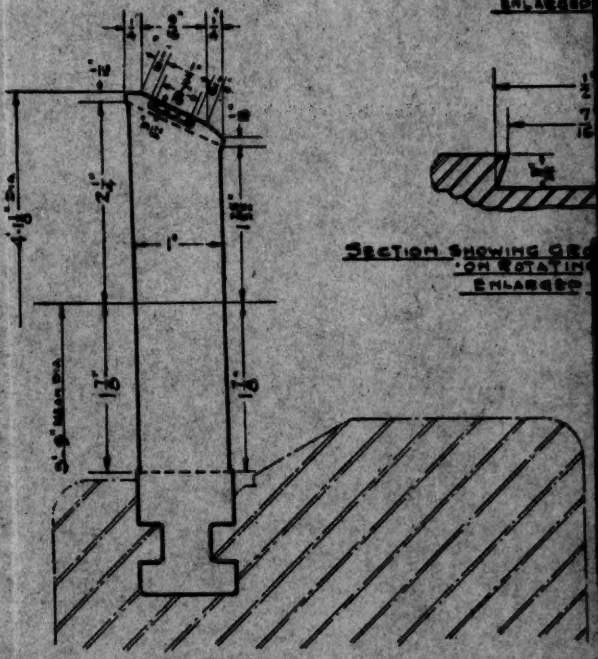
Figure 3 shows the blading, shrouding, grooves, and the method used for installing the shrouding. In order to eliminate vibration of the blades due to caulking the shrouding, it was decided to roll


FACE PLATE SECURED TO TURBINE CASING
SCREW JACK
STEEL 'V' CLAMP
WOOD BLOCK
STEEL ROLLER TO PREVENT AXIAL STRAIN
1" x 3/4" x 18 1/2" SEGMENTAL BRASS PLATE CLAMPED TO BLADES
NEW SHROUDING
3/4" DIA HARDENED STEEL ROLLER
COMPOUND TOOL REST

PLAN AT H.P AHEAD. IMPULSE WHEEL, COVER REMOVED, SHOW
OF TOOLS FOR EMBEDDING SHROUDING IN 2ND STAGE IMPULSE



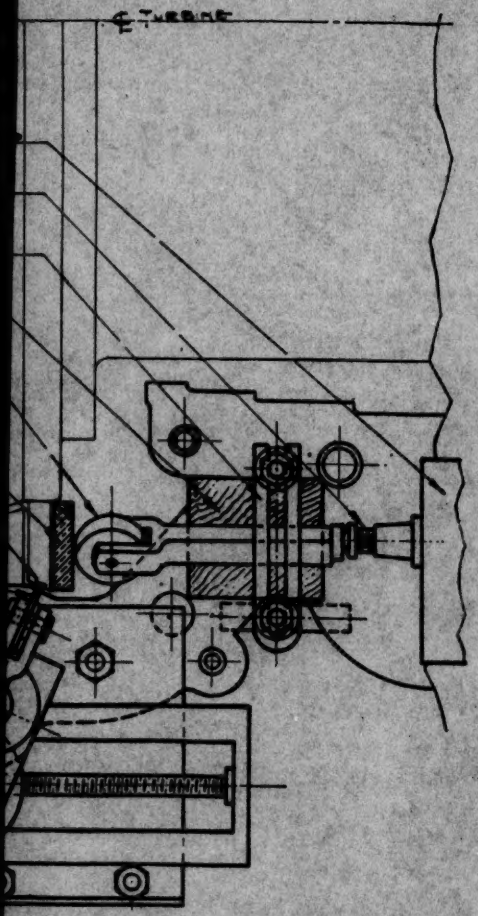
ED. SHOWING ARRGT.
MPULSE BLADES.



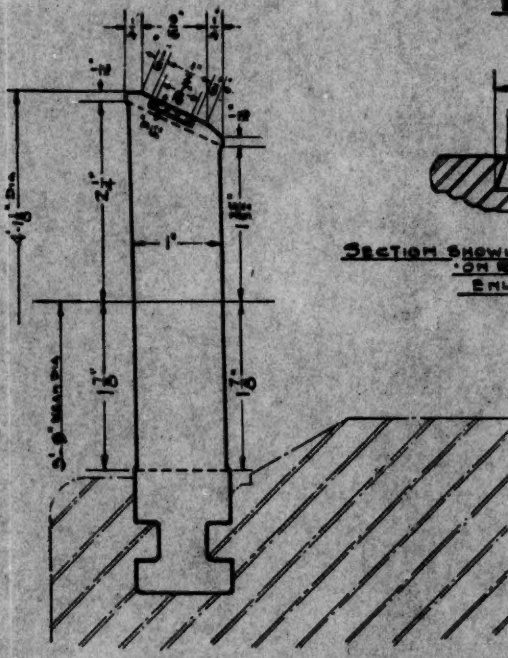
SECTION SHOWING 2ND STAGE IMPULSE BLADES
SHROUDING IN PLACE

SECTION THR
ENLARGED

SECTION SHOWING GRO
ON ROTATIN
ENLARGED

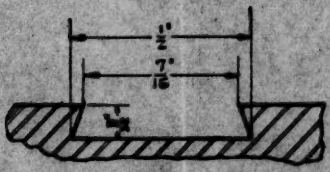



MOVED. SHOWING ARRGT.
GE IMPULSE BLADES.

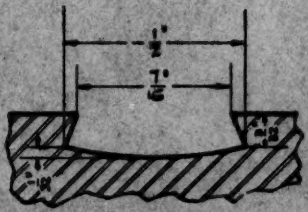



SECTION SHOWING 2ND STAGE IMPULSE BLADES SHROUDING IN PLACE

SECTION THRU SHOWING
ENLARGED 4 TIMES




OM SHOWING GROOVE FOR SHROUD STRIPS
ON ROTATING BLADES
ENLARGED 4 TIMES




SECTION SHOWING PROPOSED GROOVE FOR SHROUD
STRIPS ON ROTATING BLADES
ENLARGED 4 TIMES.

BLADE

it into the dove-tail slot. In this particular repair job, the rolling process was complicated by the fact that the heads of the blades were not flat on top, but were machined at an angle of 27 degrees, as shown in Figure 3. The same figure shows the tool used for rolling in the shrouding. It consists of two rolls, one to force and seat the shrouding in the blade groove, and the other to support the blades against side or axial thrust caused by the rolling operation. A segment 1 inch thick and 18 inches long was machined to fit against the side of the turbine blades and it was secured in place by a " C" clamp. This segment afforded a rolling surface for the side or axial which was of the flat-face, hardened type. This roll was adjustable to opposing thrust by means of a set screw in the end of the roll holder. The roll for the shrouding was mounted in the tool post of a compound rest which was secured to the horizontal joint surface of turbine casing. This roll was hardened and had a sharp radius section.

To install the shrouding, a section was placed in position in blade groove and held in place by a few light blows with a hammer. The axial roll was then set against the thrust segment, and the periphery roll against the section of shrouding previously placed; then by means of the jacking gear, the rotor was slowly revolved ahead and astern six times, feeding the roll in hard against the shrouding during the rolling process. After completing this operation, the edge of the turbine blade at the forward and after intersecting line between rotor blade and shrouding was rolled over the shrouding so as to insure a good dove-tail effect; for this purpose a hardened roller with a 27-degree face was used, as a final operation, hack saw clearance between the shroud sections was given; this is necessary as the rolling operation lengthens the shrouding.

The foregoing method proved to be entirely satisfactory. Before adapting it to use on the turbine rotor referred to experiments were made in the shop, where it was demonstrated conclusively that shrouding installed in this manner completely filled the dove-tail groove; also it was demonstrated by actual test in a special machine that the resistance to radial pull was far in excess of any possible centrifugal strains.

Figure 3 shows the shape of the dove-tail slot and the section of the shrouding actually used, a reverse curvature being given the

shrouding to make it possible to insert it in the grooves. The material as received was too thin to admit of machining the shrouding to this shape, so it was necessary to form it roughly to shape in a hydraulic press, and complete the process by machining. This is undesirable because it requires, first, forming the metal to one shape and then reverse rolling it to the other and, in addition adds to the cost and time of manufacture. This could be avoided by supplying the shroud strips sufficiently thick to permit machining to the desired thickness and radius from the bar stock.

Figure 3 shows the shape of slot actually used to be flat on the bottom; it also shows a proposed form of slot with a curved bottom; it is believed that the latter would give better results than the former. Stainless steel shrouding is very much like spring steel in its action and it is difficult to hold it in the slot during the first part of the rolling process. This would be eliminated if the arc were completely reversed on the first rolling operation, as would be the case with a curved slot; also this shape of slot would give more strength to resist centrifugal strains.

# ANALYSIS OF THE FUEL CONSUMPTION OF DIESEL MOTORSHIPS.

By C. F. Gross, Member, and B. W. Martin.

SUMMARY.

This study includes a collection of the data concerning the operation of seventy Diesel vessels which have been placed in service during the past eight years. An analysis is made on several different basis, but particularly that with regard to fuel consumption. Comparisons between the two-cycle and four-cycle are drawn with a view toward clarifying this question which is still a much disputed one in the motorship field.

## INTRODUCTION.

Transportation has been, and will continue to be one of the most important contributors to the progress of mankind. Conveyance takes place now by way of three "elements," viz.: land, water and air. In the realm of ocean transportation there has arisen a new propelling medium within comparatively recent years, viz.: the Diesel engine, the growth and use of which has been extremely rapid and widespread. Statistics show that in July, 1914, there were in service 297 motorships of 234,287 gross tons whereas in July, 1927, the number had increased to 2,552 with a gross tonnage of 4,270,824. Furthermore, the tonnage of motorships under construction today is 69 per cent of the tonnage of steam reciprocating and turbine vessels.

There are four classes of Marine Diesel engines, namely:

1. The four-cycle single acting engine.
2. The two-cycle single acting engine.
3. The four-cycle double acting engine.
4. The two-cycle double acting engine.

A knowledge of these types is presupposed, nor will mention be made of the manufactures of the various engines. Naturally, considerable controversy has arisen as to which is the better type in actual use. Manufacturers of each make claim for their own particular product and cycle. Hence, the propounding of the question: "Which shall it be, the two or the four-cycle type?" To the writers' knowledge, no attempt has been made publicly to obtain a general answer to this question. From time to time papers are presented before various marine and naval societies in which this question is debated, but because the presentor is connected with a company favoring one or the other type, the conclusions are apt to incline to the cycle with which the manufacturer is concerned. This, then, leads up to the original object of this study which was to obtain data from as many Diesel ships in operation as possible, and from an analysis of the same attempt to gain an answer to the two and four-cycle question.

The accuracy of results from such an investigation depends upon two factors:

1. A study of a large number of installations.
2. Accurate and complete data for each ship.

Unfortunately, as is often the case, both conditions were lacking to a certain degree. The sources which yielded usuable information were as follows: technical reports, papers presented to engineering societies, technical publications and correspondence with operating engineers. This made available the results of 70 ships which was a fairly good cross-section of the motorships afloat.

## COMPUTATIONS.

The mean effective pressure based on indicated or brake horsepower was computed from the dimensions of the engine. The fuel consumption was invariably reported as so many tons per day; this was converted to pounds per indicated or brake horsepower hour. The effectiveness of the burning of the fuel oil is expressed in pounds per square inch of piston area. The volume swept by the piston necessary to develop one brake horsepower per minute is a measure of the effectiveness of space utilization.

## RESULTS.

The results of the analysis of the seventy ships shows that the grand average indicated horsepower per engine was 2740 and the grand average indicated horsepower per ship was 4206. The ratio of the total installed two-cycle indicated horsepower to four-cycle indicated horsepower was 42 per cent. The average fuel consumptions are as follows:

| | |
|---|---|
| Four-cycle, pound per I.H.P. hour | 0.311 |
| Four-cycle, pound per B.H.P. hour | 0.419 |
| Four-cycle, pound per square inch of piston area | 0.191 |
| Two-cycle, pound per I.H.P. hour | 0.338 |
| Two-cycle, pound per B.H.P. hour | 0.397 |
| Two-cycle, pound per square inch of piston area | 0.295 |

Average specific volumes.

| | |
|---|---|
| Four-cycle, cubic feet per B.H.P. minute | 6.21 |
| Two-cycle, cubic feet per B.H.P. minute | 3.10 |

Average Mean effective pressures.

| | |
|---|---|
| Four-cycle, pound per square inch, based on I.H.P. | 95.9 |
| Four-cycle, pound per square inch, based on B.H.P. | 72.6 |
| Two-cycle, pound per square inch, based on I.H.P. | 97.0 |
| Two-cycle, pound per square inch, based on B.H.P. | 73.2 |

## DISCUSSION.

According to the results, the mean indicated horsepower per engine is 2740, and that per ship is approximately 4200. These two figures show that the majority of the ships analyzed were twin screw, since 4200 is considerably larger than 2740, but not double its value. This is due to the including of several large single screw installations. The reason for twin screws is well known to all readers.

The results show that over twice as much power has been built into engines using the four-cycle principle as those constructed upon the two-cycle principle. No doubt this division is fairly indicative of the whole motorship field, and may be attributed to the wide distribution of the products of Messrs. Burmeister and Wain who manufacture exclusively four-cycle Diesel engines.

About one-third of the engines installed in the ships in this investigation were manufactured by Burmeister and Wain or by their licencees.

On a basis of fuel consumption per indicated horsepower hour, the results favor the four-cycle engine which uses approximately eight per cent less fuel than the two-cycle type. When a comparison is based on the fuel consumption per brake horsepower hour it is interesting to note that the two-cycle engine requires about five per cent less fuel than the four-cycle type. These figures appear to support statements which are often made by various proponents. The comparison upon a brake horsepower hour basis is more important, as it is horsepower applied to the propeller shaft, rather than indicated horsepower in the cylinder, which drives the ship.

The results obtained for the fuel consumption per square inch of piston area were respectively: 0.191 pound for the four-cycle engines and 0.295 for the two-cycle. At first sight this would appear to favor the four-cycle group, since its average consumption is the smaller. Because of the two-cycle principle with its ensuing increased number of working strokes for a given number of revolutions, the two-cycle engine will be smaller than the four-cycle engine of the same power under like conditions. Assuming, then, that a two-cycle and a four-cycle engine have the same specific fuel consumption with equal powers, the consumption per square inch of piston area of the four-cycle will necessarily be less than that of the two-cycle because the same quantity of oil is "spread out" over a greater area; the greater area resulting from the increased size required for the four-cycle engine. Therefore, in using this item for a direct comparison, only engines of the same cross-sectional area should actually be used. Since such a condition does not hold in practice, the figures should be modified to such an extent that comparisons are made on the basis of equal areas.

The question arises as to how such modifications in the I.H.P. equals $P \times L \times A \times N \div 33{,}000$ formula can be made. Assume equal indicated horsepowers (I.H.P.); the analysis showed about equal mean effective pressures, (P); and assume equal piston speeds (LN) for a two and a four-cycle engine. N for a two-

cycle engine becomes N/2 for a four-cycle engine. But according to the conditions just made LN is to be kept the same for both engines; therefore A must be doubled for the four-cycle motor to compensate for the halving of its revolutions per minute. Furthermore, the assumption that equal piston speeds exist is not quite correct for actual engines. The analysis showed that the ratio of the piston speeds of the four and two-cycle engines were as 1.2 to 1.0 with that of the four-cycle engine the larger. It can be seen that A for the four-cycle engine may be reduced and still keep constant the portion LAN of the formula as LN is increased. Furthermore, A will be increased in the ratio that LN is increased. That is:

$$\frac{\text{Area of pistons, four-cycle engine}}{\text{Area of pistons, two-cycle engine}} \text{ equals } \frac{A(LN)}{A(LN)} \times \frac{2}{1} = \frac{1.0}{1.2}$$

or 1.67 to 1.0

Now, bringing the comparison of this section to the same basis, the figures for the four-cycle is determined with an area 1.67 times that used in the case of the two-cycle engine. Therefore the four-cycle figure of 0.191 is to be multiplied by the ratio of the areas, 1.67, because if the four-cycle area was reduced, the consumption per unit area would increase. The result is 0.319 pound of fuel burned per square inch of piston area for the four-cycle engine which is to be compared with 0.295 pound for the two-cycle engine.

Thus it is seen that what at first sight is a figure favoring the four-cycle engine actually favors the two-cycle engine when comparisons are made on the same basis. The consumption per square inch of piston area is sometimes given instead of that per indicated or brake horsepower hour, although it can be very misleading unless its real significance is understood. The particular value of a comparison figure of this type lies in the elimination of the question of indicated or brake horsepower. Such a figure conveys more information than the consumption per indicated or brake horsepower as it takes account of questions of design, namely, by incorporating the cross-sectional area of the cylinders.

The significance of the comparison of cubic feet per brake horsepower is an indicator of the size or bulk of an engine required to deliver a certain horsepower. Here the delivered horsepowers are

on the same basis for each type of engine; that is, one brake horsepower per minute. The results definitely favor the two-cycle engine, which require exactly one-half (3.10) as much volume as do the four-cycle type (6.21) to produce the same power for the same length of time. The reason is obvious. The space saved by two-cycle engines is usually estimated about 45 per cent and the weight is usually 35 to 40 per cent of a four-cycle engine. The specific volume ratio of 2 to 1 and the cross-sectional area ratio of 1.67 to 1 show how these savings of space and weight are effected in the two-cycle engine.

Higher mean effective pressures are an advantage in any engine since they result in the production of more power in cylinders of a given size. The upper limit of mean effective pressures is of course the allowable working and heat stresses in the materials of construction. It has often been claimed that the two-cycle engine has not sufficiently demonstrated that it is able to carry as high a mean effective pressure as the four-cycle engine over a long period of time. However, the analysis show that as high mean effective pressures have been used in the two-cycle as in the four-cycle engine.

No attempt was made in this study to go into the relative merits and demerits of airless injection engines as compared to those using air atomization. In such discussions, however, the statement is often made that the fuel consumption for the airless injection engine is the larger of the two. The figures from the analysis show the following:

Four-cycle airless injection type, lb. fuel per I.H.P. hour.....0.329
Four-cycle airless injection type, lb. fuel per B.H.P. hour.....0.424
Four-cycle air injection type, lb. fuel per I.H.P. hour.....0.309
Four-cycle air injection type, lb. fuel per B.H.P. hour.....0.418
Two-cycle airless injection type, lb. fuel per I.H.P. hour.....0.394
Two-cycle airless injection type, lb. fuel per B.H.P. hour.....0.408
Two-cycle air injection type, lb. fuel per I.H.P. hour.....0.331
Two-cycle air injection type, lb. fuel per B.H.P. hour.....0.394

It has already been stated that for an exact comparison all the governing conditions should be similar. That is, all the fuel consumptions should preferably be determined at normal speed and

load draft in the open sea. The ships which were used in this analysis have been taken from many sources and all conditions of tests were not reported, and therefore any deductions which we have drawn may be rightfully questioned. Nevertheless, in all comparisons it is well to remember that the installation showing the best fuel economy is not necessarily the one which will exhibit the most economical all-around performance in service. The other governing factors are: capital charges, depreciation, cost of maintenance, weight, space occupied, propeller speed and efficiency are all of equal importance, and any one may easily offset a slight advantage gained by a decreased fuel consumption.

### BRITISH MARINE OIL ENGINE TRIALS COMMITTEE.

In Great Britain, tests have been carried out upon four motorships by a committee of unbiased and competent men, the results of which have been published and are looked upon as very authoritative. These reports are extremely complete and a brief review of their plan will be given as indicative of the type of investigation which should be carried throughout the whole motorship field. This committee is composed of a chairman, deputy chairman, secretary, seven members of the Institution of Mechanical Engineers, seven members of the Institution of Naval Architects, a representative of the Admiralty and two members of the Institute of Marine Engineers. Their reports contain a description of the ship under investigation, particulars of the propeller, a general description of the engine room, particulars of the auxiliaries and deck machinery and a detailed description of the main Diesel engine.

The reports are divided into two general divisions. The first contains a description of the shore test; that is, the arrangement of the engine and apparatus and the results of the shop tests. The data taken are very complete, which includes barometric figures, fuel and lubricating oil analyses, indicator cards and their deductions. Tables contain data and information regarding injection air compressors, pressures and temperatures in the receivers and their indicator cards as well as data from the scavenge air pumps. Reduced to tabular form are brake horsepower, friction horsepower, mechanical efficiency, fuel quantities, exhaust gas observa-

tions and temperatures of cooling water. Finally a complete thermal balance is made. Curves for the shop tests also include the following: fuel consumption plotted on indicated and brake horsepower; mechanical efficiency based on mean indicated pressure and on brake equivalent mean effective pressure, fuel consumed based on speed and torque.

The second part of the report, which is likewise complete, comprises pertinent data taken during the sea trials. Finally, there is included a discussion on various phases by members of these societies, and also the result of voyages, if any have taken place. If such a similar means of testing Diesel-engined ships were adopted by our country, it would go a long way toward clearing up some of the disputed points in this field. The plan necessarily involves the friendly cooperation of the engine manufacturers, shipowners, the United States Shipping Board and our engineering societies. The British idea eliminates much of the "secrecy" regarding Diesel engines and the results are looked upon as reliable, true and unwarped by personal bias.

## CONCLUSIONS.

As a result of this investigation the following deductions have been made:

1. The fuel economy of the four-cycle engine is superior to that of the two-cycle engine on an I.H.P. basis.

2. The fuel economy of the two-cycle is slightly better than that of the four-cycle engine on a B.H.P. basis.

3. A four-cycle engine requires about twice the cylinder volume of a two-cycle engine to produce the same power.

4. The four-cycle engine has a higher fuel consumption per square inch of piston area than the two-cycle type when compared on the same horsepower basis.

5. Fuel consumptions per square inch of piston areas as usually given are misleading, as no account of the inherently increased size of the four-cycle engine is made.

6. When properly used, the fuel consumption per square inch of piston area is of more value than a comparison of fuel per indicated or brake horsepower hour, because the question of power is eliminated.

7. There is practically no difference between the average indicated and brake mean effective pressures of two and four-cycle engines.

8. The fuel consumption of airless injection engines on either horsepower basis is uniformly higher than that of the air injection type.

9. Fuel economy is not the deciding factor in the selection of a Diesel engine.

10. The four-cycle type enjoy a much wider distribution than the two-cycle engine because of priority of design and construction.

11. The majority of Diesel-powered vessels have twin screws.

12. Most manufacturers hold to a policy of not disseminating information concerning their Diesel engines.

13. Much confusion exists in the motorship field concerning the publishing of reliable data regarding motorship performance.

14. The procedure of the British Institution of Mechanical Engineers and the Institution of Naval Architects as exemplified in their Marine Oil Engine Trials Committee reports is an excellent example of the means which should be used to clarify the disputed points which arise in the testing of Diesel engines.

# MARINE BOILERS FOR HIGHER PRESSURES AND TEMPERATURES AND THEIR CONTRIBUTION TO GREATER ECONOMY.

By J. H. King,* Civil Member.

The present day demands on the machinery of ships, including both naval and merchant vessels, has led to developments in engineering that mark a new era in ship propulsion. One of the marked steps has been the trend toward higher steam pressures and temperatures, resulting in a considerable increase in operating economy.

The problems and requirements of naval vessels are quite different in many respects from the merchant vessel. However, increased efficiency and economy in the power plant is vital to both types of ships, as these result in a lower operating cost and a greater steaming radius. This latter point is of particular importance in naval vessels but is also of importance in certain types of merchant vessels from the standpoint of their use as naval auxiliaries in time of an emergency or war.

Marine engineers are necessarily more conservative in their designs than need be the case ashore. The problems of space, weight, steaming radius, reliability and accessibility for repair and also the motion of a ship at sea, are all factors that require very serious consideration. A break down of machinery in a land power plant may be inconvenient, but in a few minutes a " hook up" with some other plant may be readily made and power is then available again. Of course this is impossible at sea, where a serious break down of the ship's power plant may endanger the safety of the ship. Therefore new designs and changes must be made slowly and thoroughly proven before they can be safely incorporated in the power plant of a ship.

Editors Note—This article was prepared for the joint Washington meeting but due to the inability of Mr. King to be present, Mr. Stillman represented the Babcock and Wilcox Co., at this meeting. Although the two papers are somewhat similar both are printed in this issue.

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The practicability and economic value of higher steam pressures and temperatures have been clearly proven in land plants, as shown by the increasing use of such installations. At the present time there are, in plants ashore, more than 5,000,000 square feet of heating surface in Babcock & Wilcox boilers, alone, using pressures from 350 to 450 pounds and more than 1,250,000 square feet in boilers using pressures of from 650 to 1400 pounds.

Marine practice was formerly appreciably ahead of land practice in the use of high pressures. When 250 to 300 pounds pressure was relatively common in marine and naval practice, the standard practice ashore was to use 200 to 250 pounds pressure. The first really high pressure boiler of which the writer has knowledge was a 500-pound pressure Babcock & Wilcox Marine boiler installed at the Naval Experiment Station at Annapolis in 1915. This boiler is still in service.

The first higher pressure installation ashore was in 1915, at the Joliet Plant of the Public Service Company of northern Illinois. This installation consisted of three Babcock & Wilcox boilers of 10,000 square feet of heating surface each, built for a working pressure of 350 pounds.

In 1919 the United States Navy installed two boilers for 450 pounds working pressure, one being located at Portsmouth, Virginia, and one at Portsmouth, New Hampshire.

In 1922 the American Gas and Electric Company installed six Babcock & Wilcox boilers of 14,090 square feet of heating surface each, built for a working pressure of 650 pounds but operated at 535 pounds pressure with superheat to give a total temperature of 705 degrees F. This plant has been operating with stokers for several years under the above conditions at an average of 14,000 B.T.U. per kilowatt hour output. For the week ending April 18, 1925, this plant operated at 13,715 B.T.U. per kilowatt hour. As they use coal having a B.T.U. value of only 10,390 B.T.U. per pound as fired, this station is securing a kilowatt on 1.32 pounds of coal per hour. On the basis of 13,000 B.T.U. coal, which is more common in marine practice, the result would be 1.05 pounds of coal per kilowatt hour, which is equivalent to .78 pound of coal per shaft horsepower. Assuming the use of fuel oil containing 18,500 B.T.U. per pound, this practice would be equivalent to .55 pound of oil per shaft horsepower.

Another typical land installation that is probably of particular interest to marine people at this time on account of its use of powdered coal, is the Miami Fort Station of the Columbia Lighting and Power Company. This plant consists of six Babcock & Wilcox boilers of 15,110 square feet of heating surface each, operating at 650 pounds pressure and with superheaters to give a total temperature of 725 degrees F. This station operates at 13,000 B.T.U. per kilowatt hour output with powdered coal.

These stations indicate the economies that may be anticipated with relatively high pressures and temperatures in properly designed steam power plants.

Figures 1 and 3 show some typical high pressure Babcock & Wilcox land boilers and will indicate the general design of such boilers.

Figure 1 is one of the boilers at the Miami Fort Station referred to previously. This boiler contains 15,110 square feet of heating surface, 3184 square feet of interdeck superheating surface, 9942 square feet of economizer surface and 9888 square feet of air heater surface. The furnace volume is 12,000 cubic feet and the furnace is fitted with Bailey water-cooled walls. The relative size of furnace and boiler is quite noticeable from this cut.

## BAILEY FURNACE WALL.

### DESCRIPTION.

A general outline of the Bailey furnace wall construction is shown in Figure 2. The furnace wall blocks consist of a cast iron base formed to fit the curvature of the 3¼-inch tubes. A special tile facing of high grade refractory material, usually 1¼-inch thick, is cast into the furnace side of each block. The blocks are tightly clamped to 3¼-inch O.D. seamless water-cooling tubes, spaced on 6-inch centers, forming a continuous air-tight wall for the furnace. The refractory furnace lining thus obtained permits a high furnace temperature, but, at the same time, sufficient heat is carried back through the blocks and into the water-cooling tubes to keep the temperature of the inner or furnace surface of the wall at or below the fusing temperature of the ash in the coal being burned.

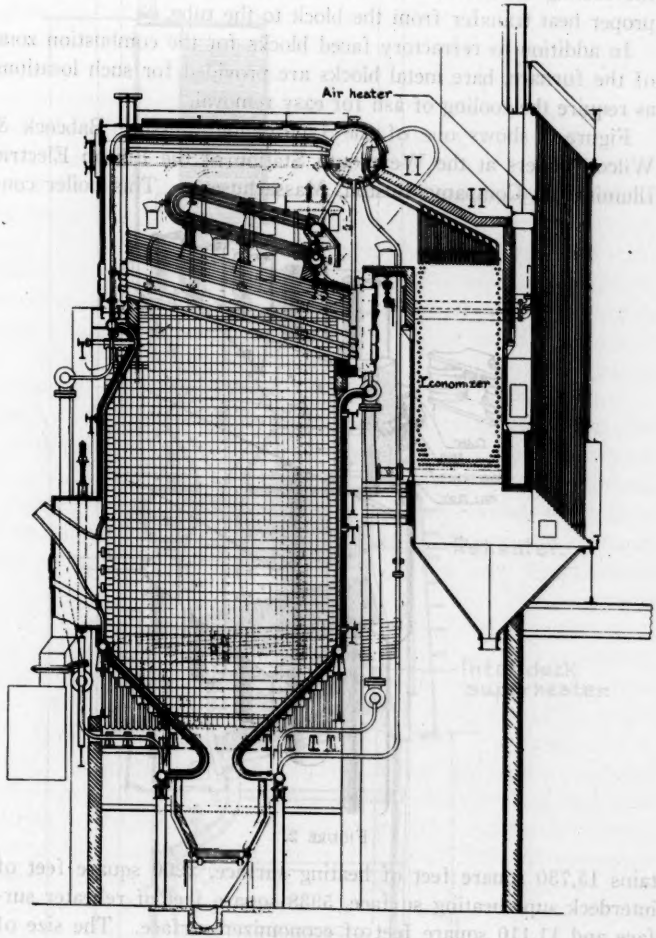



FIGURE 1.

A special heat-conducting bond is used between the block and the cooling tube, to secure the best thermal contact and provide proper heat transfer from the block to the tube.

In addition to refractory faced blocks for the combustion zone of the furnace, bare metal blocks are provided for such locations as require the cooling of ash for easy removal.

Figure 3 shows one of the 1200-pound pressure Babcock & Wilcox boilers at the Weymouth Station of the Edison Electric Illuminating Company, Boston, Massachusetts. This boiler con-

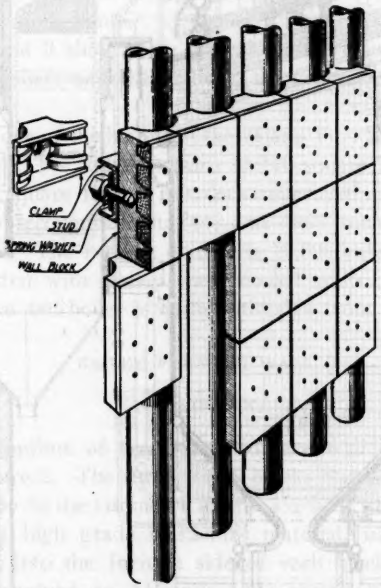



FIGURE 2.

tains 15,730 square feet of heating surface, 2200 square feet of interdeck superheating surface, 5938 square feet of reheater surface and 11,110 square feet of economizer surface. The size of the various parts of this boiler are shown in this cut.

An examination of these cuts will indicate the similarity between the design of the pressure parts of these boilers and standard Babcock & Wilcox Marine Boilers. The high pressure land boiler has

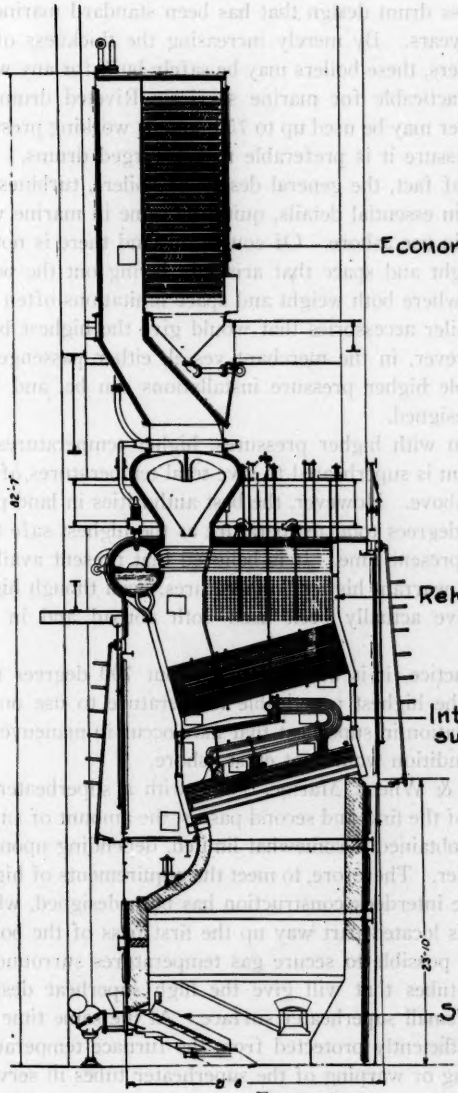

Economizer.
Reheater.
Inter deck superheater.
Stoker.
82'-3"
23'-0"

FIGURE 3.

adopted the cross drum design that has been standard marine design for many years. By merely increasing the thickness of the drums and headers, these boilers may be safely built for any working pressure practicable for marine service. Riveted drums 42 inches in diameter may be used up to 750 pounds working pressure. Beyond this pressure it is preferable to use forged drums.

As a matter of fact, the general design of boilers, turbines and auxiliaries are, in essential details, quite the same in marine work as that already in use ashore. Of course on land there is not the problem of weight and space that arises in laying out the power plant of a ship, where both weight and space limitations often preclude certain boiler accessories that would give the highest boiler economy. However, in the merchant vessel, either passenger or freighter, suitable higher pressure installations can be, and have already been, designed.

In conjunction with higher pressures, higher temperatures result and the steam is superheated to give total temperatures of 750 degrees F. and above. However, the best authorities in land practice regard 750 degrees total temperature as the highest safe temperature at the present time. It is believed that present available materials do not warrant higher temperatures, even though higher temperatures have actually been used both abroad and in this country.

In marine practice, it is believed that about 700 degrees total temperature is the highest practicable temperature to use on account of the variation in superheat that may occur in maneuvering a ship. This condition would not exist ashore.

In a Babcock & Wilcox Marine Boiler with a superheater located at the top of the first and second passes, the amount of superheat that can be obtained is somewhat limited, depending upon the height of the boiler. Therefore, to meet the requirements of higher temperatures, the interdeck construction has been designed, where the superheater is located part way up the first pass of the boiler. In this way it is possible to secure gas temperatures surrounding the superheater tubes that will give the high superheat desired with a relatively small superheater surface. At the same time the superheater is sufficiently protected from the furnace temperature to prevent burning or warping of the superheater tubes in service.

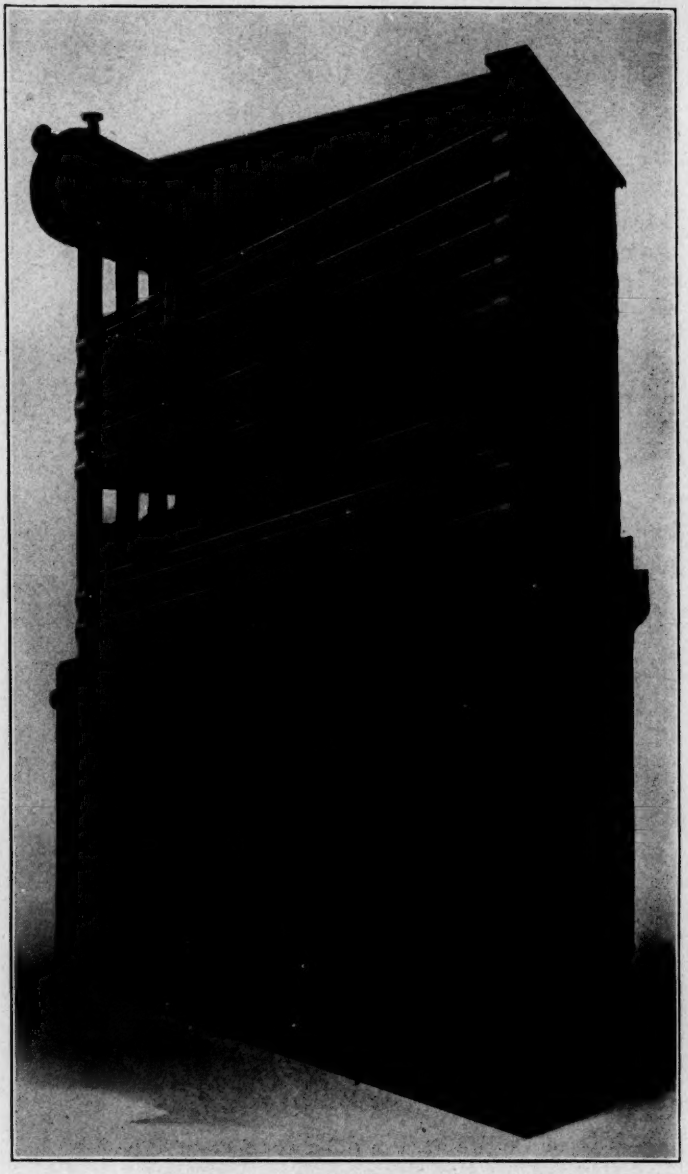


FIGURE 4.

One of a Battery of Two Babcock & Wilcox Boilers, with Interdeck Superheaters and Air Heaters, Built for the Steamer *Carl D. Bradley* of the Bradley Transportation Company, Rogers City, Michigan. Furnaces Arranged for Mechanical Stokers. View shows Boiler Partly Erected.

This is particularly necessary in marine practice on account of the varying quantities of steam that pass through the superheater when a ship is maneuvering, such as when docking.

Some years ago, the Babcock & Wilcox Company conducted experiments at its Bayonne plant with a radiant type of superheater in a standard Babcock & Wilcox marine boiler. As a result of these experiments, it was concluded that if the superheater was sufficiently exposed to the furnace to give the high superheat desired, it was very apt to burn out under maneuvering conditions. If the superheater were sufficiently protected by means of refractories or screening, to insure a normal operating life while maneuvering, it was impossible to secure the high superheat desired with the small furnaces available in marine practice. Similar results were obtained in experiments conducted by the Navy at the Fuel Oil Testing Plant. Interdeck design superheaters are therefore essential in marine work, where high superheat is required.

Figure 4 shows a boiler partially erected, showing the location of an interdeck superheater. This boiler is one of the Babcock & Wilcox boilers, fitted with underfeed stokers, installed on the S. S. *Carl D. Bradley.*

In discussing higher pressures and temperatures, the question is occasionally raised as to the necessity for higher pressure if lower pressure steam is available at any desired high temperature. Frequently the point has been raised that if the total B.T.U. in a pound of low pressure steam is as high as the total B.T.U. in a pound of higher pressure steam, the same economies should be secured provided both turbines are exhausting into the same vacuum. This is not the case, as readily shown by the following tabulation, which assumes the use of a 100 per cent efficient turbine in extracting the available heat in each case. The back pressure is assumed at 2 inches of mercury:

| | | | | |
|---|---|---|---|---|
| Superheat, degrees F | 213.5 | 232.2 | 270 | 310 |
| Steam Pressure (absolute), pounds | 600 | 400 | 220 | 220 |
| Steam Temperature, degrees | 700 | 677 | 660 | 700 |
| Total heat 1 pound steam (B. T. U.) | 1,350 | 1,350 | 1,350 | 1,372 |
| Heat in steam at 2 inches (B. T. U.) | 885 | 910 | 945 | 955 |
| Heat given up to turbine (B. T. U.) | 465 | 440 | 405 | 417 |
| Percent moisture in the steam entering condenser, per cent | 21 | 18.7 | 15.4 | 14.3 |

From the above it may be noted that the 600-pound 700-degree installation delivered about 60 more B.T.U. to the turbines than either 220-pound installation, this being an increase of nearly 15 per cent in the heat absorbed per pound of steam by the 600-pound installation as compared to that absorbed by the 220-pound installation.

Actually, due to friction losses in the turbines, etc., which are fairly uniform for all three pressures, the figures above are not secured in service, although relatively, the difference between them remains fairly constant. As an illustration of this difference, instead of securing 21 per cent moisture in the steam going to the condenser from a 600-pound 700-degree installation, the moisture will be nearer 11½ per cent, and in each case the theoretical percentage of moisture is approximately cut in half due to this friction. In a way this is fortunate, as it is not customary to try to operate turbines with much in excess of 15 per cent moisture in the steam to the condenser. If this condition exists, it is usually desirable to introduce a reheat cycle somewhere in the system before the condenser is reached.

It will be noted from the above that pressures in excess of 600 pounds will require reheating at some point in the steam cycle and would be perhaps desirable at a somewhat lower pressure on account of the moisture actually present in the steam. Although reheating in the boiler is quite common practice ashore, it is not deemed practicable in marine installations, at least at present, on account of the weight and space complications that would result. For that reason it is recommended that marine practice not exceed about 550 pounds pressure at the present time.

For the benefit of those readers who are not familiar with land practice, it may be well to explain here what is meant by "reheating." Very high pressure steam, in passing through the various stages of the turbine, reaches a point where the steam still has considerable pressure, but has lost all or a large part of its superheat. This steam is then by-passed from the turbine and passes back to the boiler and through the reheater. This reheater really is, in effect, another superheater. After passing through the reheater, the steam having regained its temperature, again returns to other stages of the turbine, or, in some cases, another separate turbine.

As boiler pressures are increased, the temperature of the tubes rises and the exit gas temperatures correspondingly increase. This results in a loss of boiler efficiency unless some means are taken to overcome this loss. In practically all high pressure plants ashore, either economizers or air heaters are installed in order to recover the heat in these higher uptake gases.

The benefit of recovering this heat is quite evident from the fact that every 40 degrees F. reduction in uptake temperature is equivalent to an increase of 1 per cent in efficiency.

Although several million square feet of economizer surface have been installed in shore stations by the Babcock & Wilcox Company, nevertheless it is believed that economizers are not practicable in marine work, for the following reasons:

It is necessary to use steel tubes in economizers, because of the higher pressures at which modern installations are operating. Steel tubes are subject to very rapid corrosion unless the dissolved oxygen in the feed water is reduced to not exceed 1/10 cubic centimeter per gallon. This condition is practically impossible to secure unless a proper de-aeration system is installed in the ship, as is almost always the case ashore. However, the weight and bulk of these de-aerators render them entirely impracticable on ships, as they would take up as much room and weight as a boiler itself.

With a closed feed system, the air in the feed water will be somewhat decreased, but not sufficiently so as to prevent corrosion, as it is almost impossible to keep the system tight. Consideration has been given to increasing the velocity of the water in economizers to a point where the air in the water would not have an opportunity to attack the steel. Experiments have been conducted which indicate that this theory is not entirely true or feasible. Also in marine work the boilers are operating under port conditions so much of the time that it is not possible to maintain a high velocity in an economizer. Furthermore, an economizer introduces a large number of joints and pressure parts requiring attention, in a part of the ship that is least accessible for overhaul and repair.

Air heaters, on the other hand, offer a practicable method of reducing uptake temperatures and thereby increasing efficiency.

An important advantage of an air heater over an economizer is that the preheated air secured with an air heater is very valuable in improving combustion conditions obtaining in the small marine furnace. The smaller the furnace the more important it is that the speed of combustion be increased as much as possible. Preheated air assists in doing this very effectively, regardless of whether the fuel is coal or oil. With the high grade refractories now available the effect of increased temperature on the brickwork is negligible.

Air heaters can be installed on top of the boilers or in the uptake. A very compact and light weight arrangement of air heater is shown in Figure 5. This heater consists of horizontal tubes located on top of the boiler. The air passes through the inside of the tubes and the gases make three passes over the tubes. The hot air ducts lead down the back of the boiler and under the furnace floor to the furnace front. In this arrangement the rear boiler casing is set out so that the boiler itself forms one side of the duct and by means of a door, access may be had to the rear headers and hand hole plates when the boiler is not in use.

Fans are usually required to handle this heated air, but there are so many such fans in service that this problem has been satisfactorily solved. One convenient arrangement consists of a small geared turbine-driven fan that requires very little attention and is not affected by temperature.

Increased pressures and boiler ratings ashore have led to the adoption of water-cooled furnace walls. It has been customary for many years with Babcock & Wilcox Marine Boilers to use water-cooled side walls (side boxes) in hand-fired coal installations. It is not believed desirable, however, in the small marine furnace, to use water-cooled walls with oil fuel, pulverized coal or stokers. The refractory lining of the small marine furnace is a very great aid to combustion and the cooling effect of water-cooled walls would seriously hinder the rapid combustion essential to high boiler efficiency. The very high grade refractories now available also make it unnecessary to use water-cooled walls in order to prolong the life of the firebrick, especially with oil-fired boilers.

When burning coal, either powdered or with stokers, it is sometimes desirable to use water-cooled walls in order to prevent slag

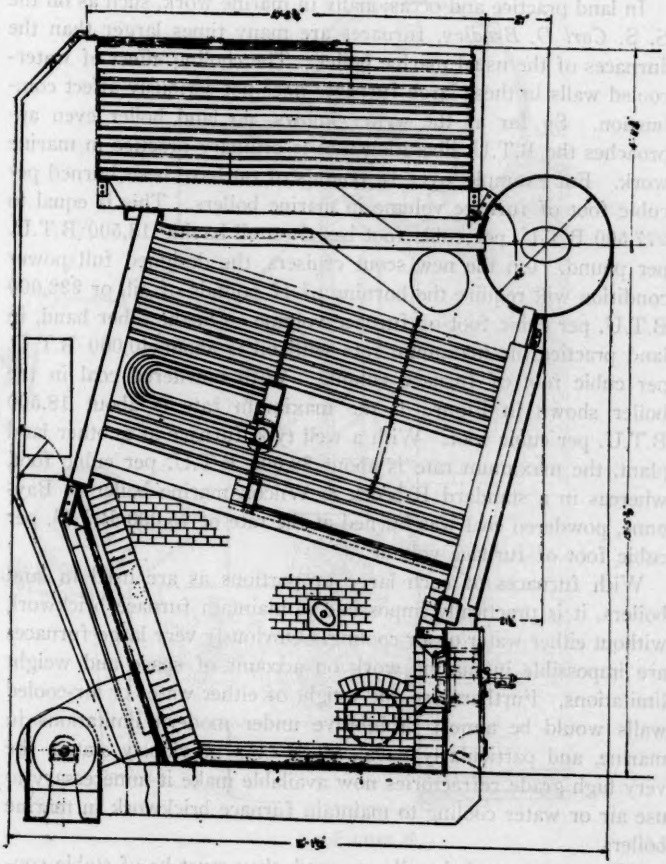


FIGURE 5.

formation with certain grades of coal. In such cases ample circulation must be provided and careful attention given to the quality of the water, as scale formation in the cooling surface would cause considerable trouble.

In land practice and occasionally in marine work, such as on the S. S. *Carl D. Bradley,* furnaces are many times larger than the furnaces of the usual marine boiler. The cooling effect of water-cooled walls in these large furnaces does not seriously affect combustion. So far as the writer knows, no land boiler even approaches the B.T.U. liberation that is ordinary practice in marine work. For example, over 15 pounds of oil have been burned per cubic foot of furnace volume in marine boilers. This is equal to 277,500 B.T.U. per cubic foot based on oil having 18,500 B.T.U. per pound. On the new scout cruisers, the designed full power condition will require the burning of 12 pounds of oil, or 222,000 B.T.U. per cubic foot of furnace volume. On the other hand, in land practice the maximum rate with oil is about 40,000 B.T.U. per cubic foot of furnace volume. With powdered coal in the boiler shown in Figure 1, the maximum rate is about 18,500 B.T.U. per cubic foot. With a well type furnace at another land plant, the maximum rate is about 38,000 B.T.U. per cubic foot, whereas in a standard Babcock & Wilcox marine boiler at Bayonne, powdered coal was burned at the rate of 75,000 B.T.U. per cubic foot of furnace volume.

With furnaces of such large proportions as are used in land boilers, it is practically impossible to maintain furnace brickwork without either water or air cooling. Obviously very large furnaces are impossible in marine work on account of space and weight limitations. Furthermore, the weight of either water or air-cooled walls would be almost prohibitive under modern limitations in marine, and particularly naval, work. As previously stated, the very high grade refractories now available make it unnecessary to use air or water cooling to maintain furnace brickwork in marine boilers.

Where water-cooled walls are used, they must be of stable construction and should be protected by refractory material. Such a construction will add considerable weight to the boiler.

In naval work, higher pressures and temperatures present quite another problem. In the Navy, one of the *primary* considerations

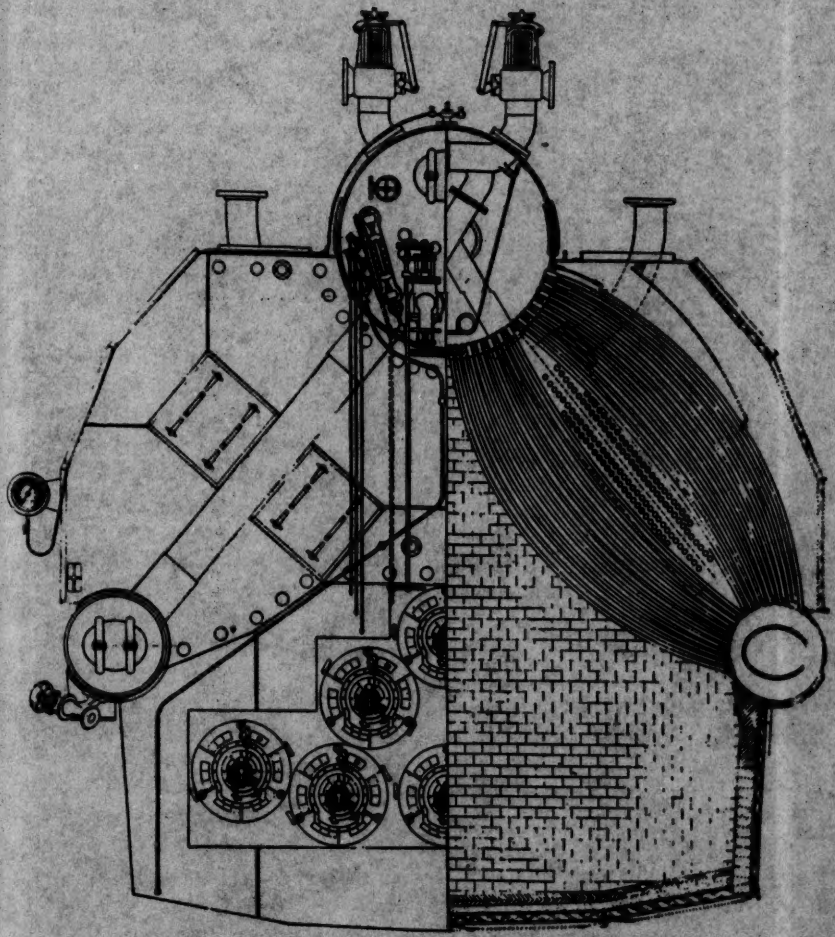
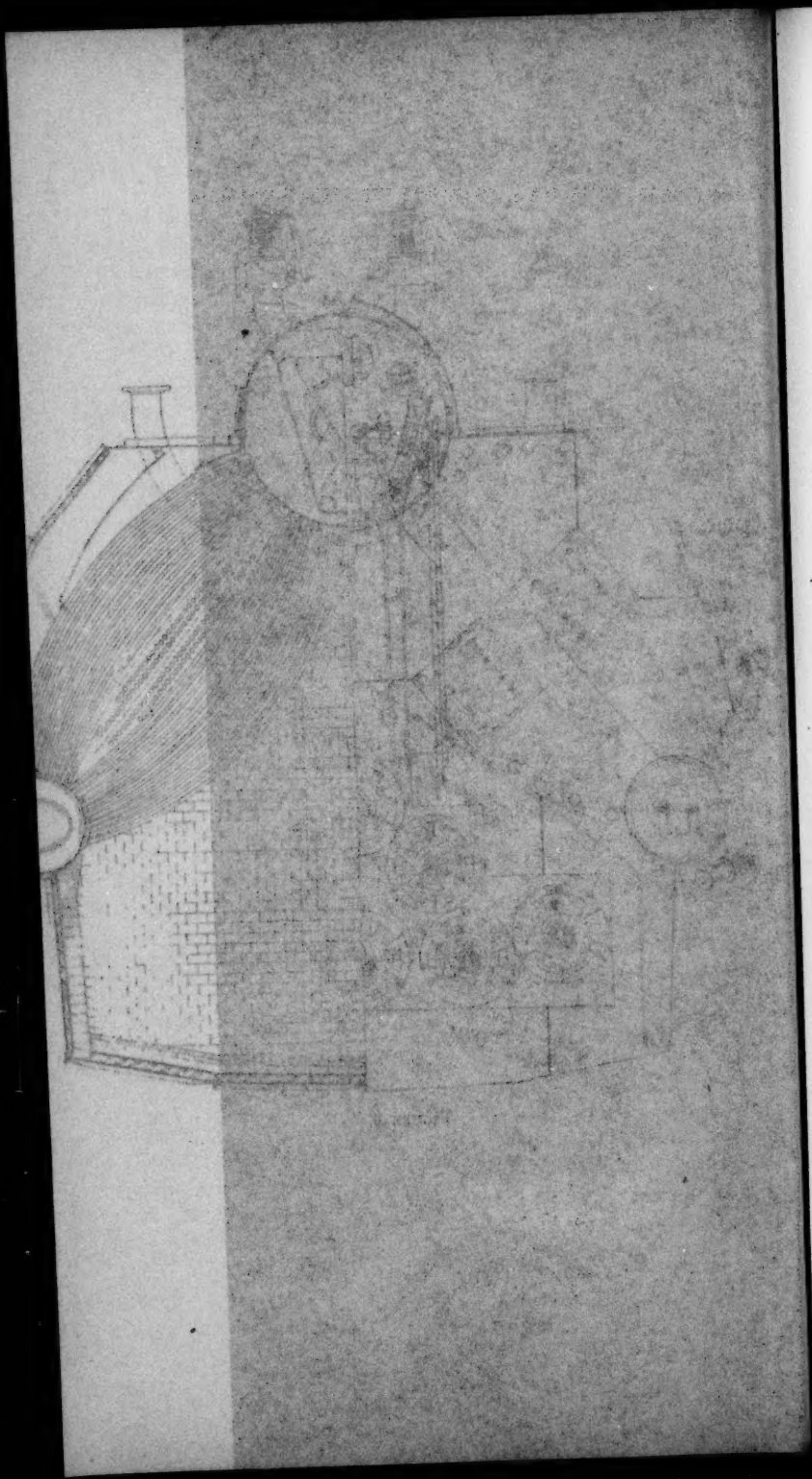


FIGURE 6.



is weight. This is especially true now with the limitations in weight set by the disarmament treaty. Lack of space is also a problem and therefore, both on account of weight and space, it is not possible for the naval boiler to use either economizers or air heaters to reduce the stack temperature that would result with higher steam pressures and temperatures.

Requirements of weight and space have led to the adoption of the small tube Express Type boiler for naval use. This seems both wise and practicable where very high loads are required for short periods and a comparatively light load is ordinarily used.

Figure 6 shows a Babcock & Wilcox Express Type Boiler (White Forster) with a superheater located at the center of each tube bank. This boiler is similar to the boilers installed on the airplane carrier *Saratoga* and the modernized battleships *Arkansas, Florida, Utah,* and *Wyoming.*

In the usual Express Type boiler, an increase in pressure will result either in an increase in drum thickness or an increase in the ligament between tubes, or both. If the drum thickness is increased very much, the weight of the boiler increases and for even a moderate increase in pressure, the plate required may not be practicable to use or even manufacture. With an increase in the ligament, the tube spacing is such that the boiler is not as good a heat strainer and so the boiler efficiency may drop off more than any possible gain that may be obtained by the turbine. Therefore, with the limitations of weight and space required for the Navy, the present Express Type of boiler practically prohibits higher steam pressures in naval vessels.

The solution of this problem lies in some new design of boiler that will be light, compact and give a very high boiler efficiency with a resulting low stack temperature. The Babcock & Wilcox Company has been working on this problem for a number of years and expects to be in a position to offer such a boiler in the very near future.

In merchant work, however, higher pressures and temperatures are not only practicable at the present time, but are also commercially sound. Several private steamship companies have already made a start and have ships either built or building. These may be regarded as the first step toward higher pressures and temperatures in marine work.

The S. S. *Malolo,* owned by the Matson Navigation Company and one of the last products of Cramps shipyard, is making a very fine record in fuel economy. This ship has 27,000 shaft horsepower and is equipped with Parsons Turbines and DeLaval Reduction Gears. There are twelve Babcock & Wilcox oil-fired boilers built for 275 pounds working pressure with superheaters located on top of the boilers to give 125 degrees F. superheat. The expected performance of this installation was .8 pound of oil per shaft horsepower with a boiler efficiency of 80 per cent. In actual service, these results have been exceeded and owing partly to a higher superheat than anticipated and partly to exceptionally well-designed auxiliaries, the ship is operating on approximately .76 pound of oil per shaft horsepower.

The S. S. *California* of the Panama Pacific Line (International Mercantile Marine) is another ship that has recently been placed in service that has exceeded its anticipated performance. This ship is electrically driven with General Electric Turbines and Motors and is designed for 17,000 shaft horsepower. Most of the auxiliaries are also electrically driven. This ship is fitted with 12 Babcock & Wilcox oil-fired boilers built for 275 pounds working pressure with superheaters on top of the boilers to give 110 degrees F. superheat. It was expected that the performance of this ship would be .802 pound of oil per shaft horsepower for all purposes, with 82 per cent boiler efficiency, but these results have actually been bettered in service.

The S. S. *Carl D. Bradley,* owned by the Bradley Transportation Company, is a self-unloading ore carrier operating on the Great Lakes. It is rather unusual in that the boilers have high furnace settings and are equipped with Westinghouse Underfeed Stokers. This ship has developed 4878 shaft horsepower on 1.05 pounds of coal for all purposes with a combined boiler efficiency of 83.9 per cent. It is fitted with Babcock & Wilcox boilers built for a working pressure of 325 pounds, with interdeck superheaters to give 290 degrees F. superheat. Air heaters are also fitted on top of the boilers.

The Morgan Line passenger steamer *Dixie,* owned by the Southern Pacific Steamship Company, is also making an excellent record in service. This ship has turbine gear drive developing 8000 shaft

horsepower and is fitted with oil-fired Babcock & Wilcox boilers built for a working pressure of 350 pounds, with interdeck superheaters to give 200 degrees F. superheat. Air heaters were fitted in the uptakes by the shipyard. This ship was designed to operate at .8 pound of oil per shaft horsepower, all purposes, but is lowering this rate in actual service.

Mr. Theodore E. Ferris, Naval Architect, New York, has designed a 6500-ton collier to develop 2900 shaft horsepower. This ship will have geared turbines and Babcock & Wilcox boilers designed for a working pressure of 400 pounds, with interdeck superheaters to give 250 degrees F. superheat. Air heaters are to be fitted on top of the boilers. This ship is rather unusual in that it will burn pulverized coal. It is expected that the coal per shaft horsepower for all purposes will be 1.02 pounds.

A proposal has been made to the Shipping Board to equip a 3300 shaft horsepower freighter with higher pressure geared turbines and boilers. This ship would operate with boilers built for a working pressure of 400 pounds, equipped with interdeck superheaters to give 230 degrees F. superheat and with air heaters located on top of the boilers. Builders are prepared to guarantee an oil rate of .63 pound of oil per shaft horsepower for all purposes. The relatively low investment cost and the low operating and maintenance cost of such an installation offers a bright future for merchant ships.

In general, with higher pressures and temperatures, boilers will save considerable space and weight over lower pressure installations and give a marked increase in economy. Due to the lower water rate of the turbine, the quantity of steam required is considerably decreased. Therefore less boiler heating surface is required, which results in a smaller boiler room and allows increased space for either fuel or cargo.

For example, the International Mercantile Marine is now building a new 17,000 S.H.P. ship, the *Virginia,* which is to be a sister ship of the new *California,* recently placed in service. The *Virginia,* however, will use higher pressures and temperatures. The steam requirements are thereby reduced, owing to greater economy in the turbine, with the result that the *Virginia* will have only eight Babcock & Wilcox boilers as compared to the twelve

boilers on the *California*. The boilers for the *Virginia* actually cost less money than the boilers on the *California* on account of the lesser heating surface. The turbines on the *Virginia* cost just the same as the *California's* turbines, so that the actual investment cost of the boilers and turbines on the *Virginia* is lower than on the *California*. Yet, the *Virginia* will have a decided increase in economy and therefore a lower operating cost.

The preceding paragraphs present briefly the present trend in steam practice. It is believed that it will therefore be evident that higher pressures and temperatures for marine service are not experimental and that these higher pressures and temperatures will result in a marked decrease in fuel consumption due to the higher turbine and combined boiler economy. The upkeep and maintenance cost will be as low or lower than the present marine plant. Therefore these advantages, together with the comparatively low investment cost, offer an entirely sound and commercially practicable solution to our marine problem.

**FEBRUARY 10 (Friday) —8 P. M.— COSMOS CLUB HALL**

# Joint Meeting

WASHINGTON SECTION

## THE AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

# STEAM

## UTILIZATION

**1. Higher Steam Pressures**
MR. N. E. FUNK, *Asst. Chief Engineer,*
Philadelphia Electric Co., Phila., Pa.

**5. Turbines for High Pressures**
REPRESENTATIVES FROM GENERAL ELEC. CO.,
AND WESTINGHOUSE ELEC. & MFG. CO.

—— DISCUSSION ——

DR. M. G. LLOYD, *Chairman.*
H. E. BRADLEY, *Sec'y-Treas.*

## THE AMERICAN SOCIETY OF NAVAL ENGINEERS

# STEAM

## MARINE APPLICATION

**3. Recent Experiences with Powdered Coal on S. S. "Mercer."**
COMDR. J. S. EVANS, U. S. N.,
Fuel Testing Plant, Phila., Pa.
MR. C. J. JEFFERSON OF U. S. FLEET CORP.
CAPT. R. D. GATEWOOD (C. C.) U. S. N.
of U. S. Fleet Corp.

**6. Higher Pressures for Marine Use**
C. P. WETHERBEE, *Marine Dept.*
Westinghouse Electric & Mfg. Co.

REAR ADMIRAL JOHN HALLIGAN, U.S.N., *President.*
COMDR. H. B. HIRD, U.S.N., *Sec'y-Treas.*

WASHINGTON SECTION

## THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

# STEAM

## GENERATION

**2. The Use of Pulverized Fuels**
MR. HENRY KREISINGER, *Research Engineer,*
Combustion Engin'g Corp., New York.

**4. Marine Boilers for High Pressures**
MR. J. H. KING, *Marine Dept.*
The Babcock & Wilcox Co., New York.

—— DISCUSSION ——

E. C. MAGDEBURGER, *Chairman.*
L. J. ROSE, *Sec'y-Treas.*

# HIGHER STEAM PRESSURES.*

By N. E. Funk, Manager and Chief Engineer,
Operating and Engineering Department,
Philadelphia Electric Company.

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The purpose of this paper is to deal with the subject of "Higher Steam Pressures" more in line of the economic limitations of the use of the high pressure steam itself and the conditions that must exist to justify that use, than in the line of the development of materials and apparatus for its utilization. The past few years have seen a very considerable improvement in station efficiencies; accomplished by improvements in design of apparatus, development of steel-tube economizers, air preheaters, water-walls, deaerators, evaporators, etc., adoption of reheating and regenerative cycles and the use of higher steam pressures and temperatures; and, in some cases, the new use of other heat conversion mediums such as the Mercury Vapor Process.

## TREND OF POWER GENERATION DEVELOPMENT.

The trend of improvement is illustrated in Figure 1. The steam cycle curve is based on the typical performance reports of 22 generating stations of 60,000 K.W. capacity or greater plotted against the date of installation. The flat portion of this curve from 1916 to 1922 is indicative of the retardation of power station development in the period of stress during and following the war. The marked drop after 1922 shows the effect of increasing efficiencies to combat the high coal prices of the post war period. The dotted portion of the steam cycle curve shows an estimate of the future possibilities of the straight steam cycle. A similar estimate for the mercury vapor-steam cycle is also shown. The line showing the B.T.U. in one K.W.H. is drawn to illustrate the wide mar-

* Paper read before a joint meeting of the Washington Sections of the A. I. E. E., A. S. M. E., and A. S. N. E.

gin between the past and present performance and the theoretically perfect conversion of the heat in the fuel. This margin is, of course, largely due to the limitations of the cycles and mediums used and in the present stage of the art a close approach to the theoretical seems rather remote despite the great improvements from 1913 to date.

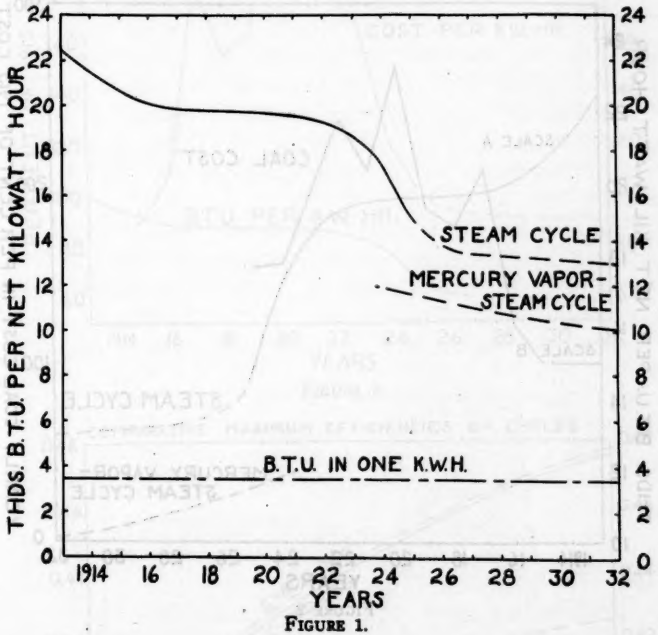



FIGURE 1.

A comparison of the increase in station economy and the increase in coal cost is shown in Figure 2. The coal costs are average yearly values for bituminous coals taken from government reports and for purposes of comparison are expressed in per cent of the 1913 cost. In view of the present strife in the coal industry it is hardly reasonable to expect any considerable reduction in the cost of coal for the immediate future.

Figure 3 shows the change in B.T.U. per K.W.H. and the change in coal cost per K.W.H. expressed in per cent of the

1913 value. These curves show particularly the excessive post war costs responsible for the present rate of improvement. It is interesting to note that, although the B.T.U. rate in 1925 had decreased to 65 per cent of the 1913 value, the cost per K.W.H. was practically the same as in 1913.

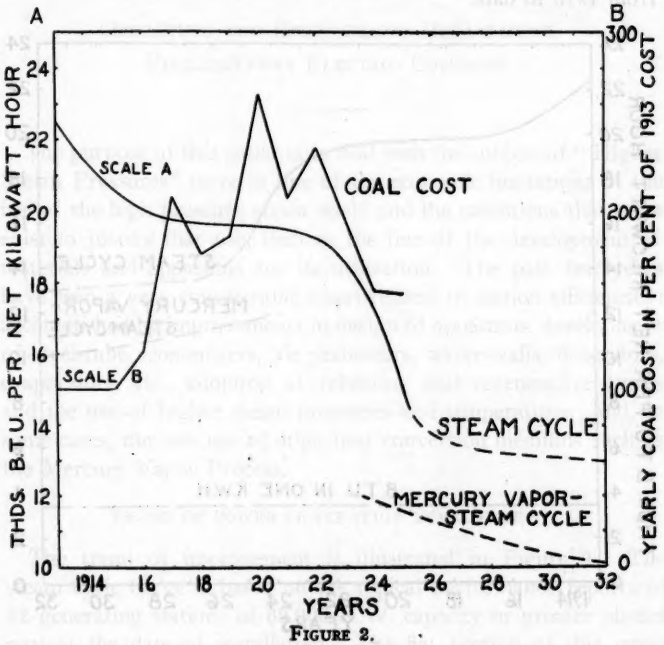



FIGURE 2.

CYCLE EFFICIENCIES.

The effect of higher pressures and temperatures with various steam cycles can be seen by reference to Figure 4. The cycle efficiency, of the three cycles indicated, increases appreciably with higher pressures, but more especially in the case of the Renegerative cycle. At 200 pounds pressure the cycle efficiency of the Regenerative cycle is 3 per cent more than the Rankine; while at 1200 pounds pressure the efficiency is 7 per cent greater with the Regenerative cycle. Increasing the temperature of the steam

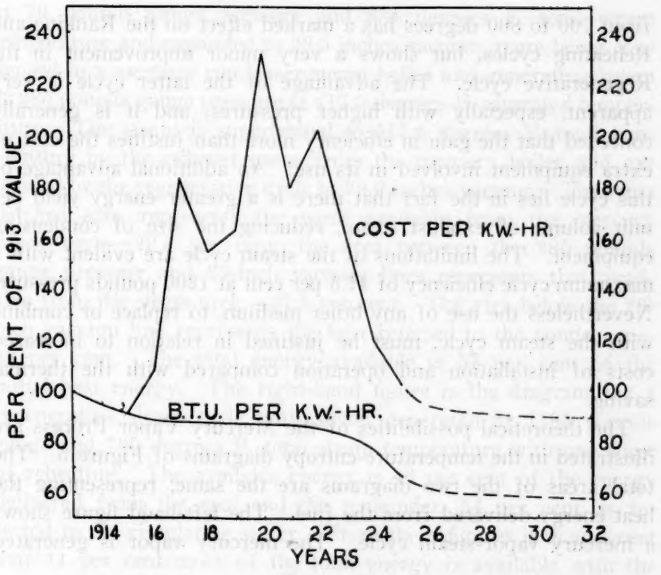



FIGURE 3.

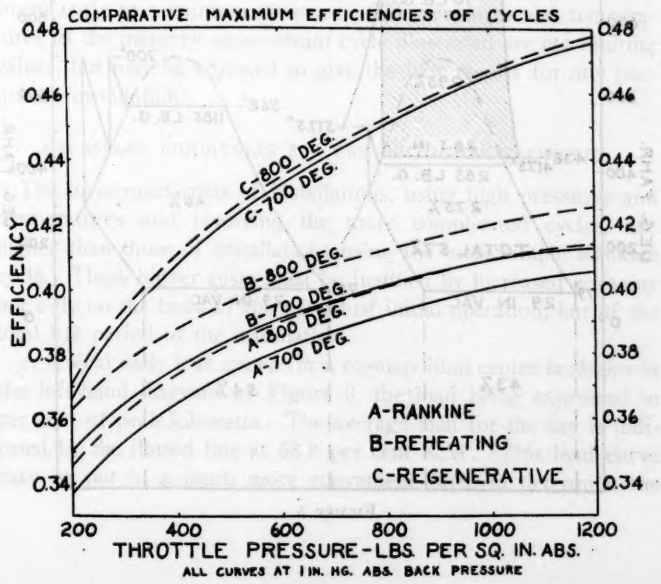



FIGURE 4.

from 700 to 800 degrees has a marked effect on the Rankine and Reheating cycles, but shows a very minor improvement in the Regenerative cycle. The advantage of the latter cycle is very apparent, especially with higher pressures, and it is generally conceded that the gain in efficiency more than justifies the cost of extra equipment involved in its use. An additional advantage of this cycle lies in the fact that there is a greater energy yield per unit volume of exhaust steam, reducing the size of condensing equipment. The limitations of the steam cycle are evident with a maximum cycle efficiency of 47.5 per cent at 1200 pounds pressure. Nevertheless the use of any other medium, to replace or combine with the steam cycle, must be justified in relation to increased costs of installation and operation compared with the thermal saving.

The theoretical possibilities of the Mercury Vapor Process are illustrated in the temperature-entropy diagrams of Figure 5. The total areas of the two diagrams are the same; representing the heat energy delivered from the fuel. The left-hand figure shows a mercury vapor-steam cycle. The mercury vapor is generated

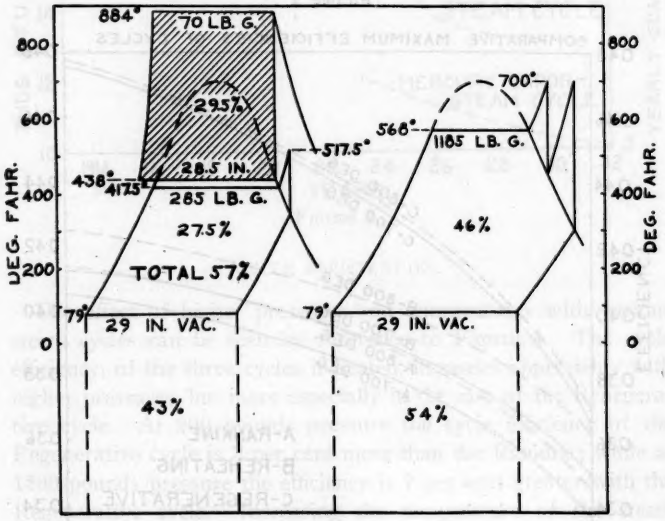



FIGURE 5.

at 70 pounds gauge pressure and 884 degrees F. total steam temperature and expanded to 28.5 inches vacuum, there being condensed in a mercury condenser-steam boiler and generating steam at 285 pounds gauge pressure at 417.5 degrees F. saturated temperature. The steam is superheated to 517.5 degrees F. total temperature by the exhaust gases from the mercury boiler and expanded by the regenerative cycle to 29.0 inches vacuum. The cross hatched area represents the work available from the mercury vapor cycle—29.5 per cent; the area between the 285 pounds gauge pressure and 29-inch vacuum lines represents that available from the steam cycle—27.5 per cent. The area below the 29-inch vacuum line represents the heat rejected to the condenser—43 per cent. The total energy available is 57 per cent of the initial heat energy. The right-hand figure is the diagram for a regenerative steam cycle with steam generated at 1185 pounds gauge and 700 degrees F. total steam temperature with one stage of reheating. The available energy is 46 per cent of the energy delivered from the coal and the remainder, 54 per cent, is rejected to the circulating water. From these figures it is apparent that 11 per cent more of the total energy is available with the mercury vapor-steam cycle shown than with the regenerative steam cycle at maximum pressure. The pressures and temperatures of the mercury vapor-steam cycle illustrated are not limiting values, but may be adjusted to give the best results for any particular installation.

## NECESSARY CONDITIONS FOR USE OF HIGHER PRESSURES.

The investment costs of installations, using high pressures and temperatures and involving the more complicated cycles, are higher than those of installations using the more simple Rankine cycle. These higher costs must be justified by increased economy not only on the basis of the period of initial operation, but of the total life period of the installation.

A typical daily load curve for a cosmopolitan center is shown in the left-hand diagram of Figure 6, the load being expressed in per cent of peak kilowatts. The average load for the day is indicated by the dotted line at 58.8 per cent K.W. This load curve may be put in a much more convenient form to determine the

possible loading of the various generating stations of a large system. This new form is shown in the occurrence curve in the right-hand diagram. The ordinate of both figures is the same, but in the second diagram the abscissa is per cent of K.W.H. generated. For each ordinate value there is plotted, as abscissa, the per cent of day's total K.W.H. generated at or below that K.W. value as indicated by the dotted lines: i.e., 100 per cent K.W.H. are generated at or below 100 per cent K.W., 86 per cent K.W.H. are generated at or below 58.8 per cent K.W., etc. The following may be an aid in understanding the occurrence curve. If the day's total K.W.H. had been generated at 100 per cent load factor,

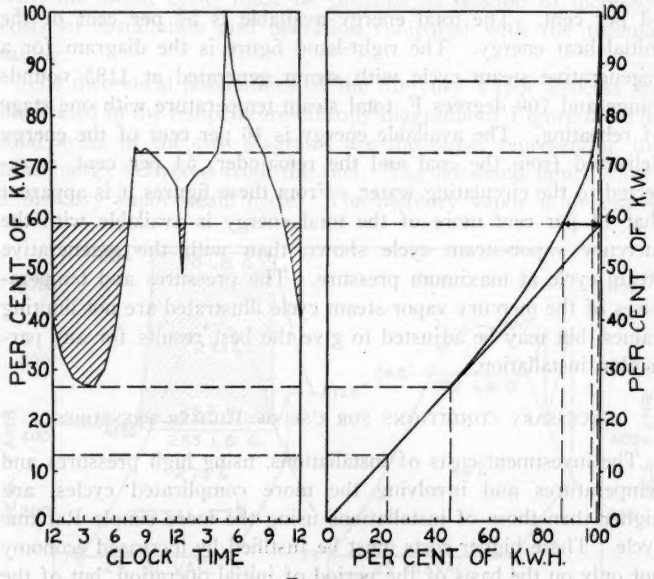



FIGURE 6.

the load curve would be represented by the complete block below 58.8 per cent K.W. in the left-hand diagram, including the shaded area. The occurrence curve would then have been the straight or 100 per cent load factor line extending from 0 per cent K.W.H. to 58.8 per cent K.W. in the right-hand diagram. However, in

the actual condition, this shaded area was not generated; causing the occurrence curve to fall away from the 100 per cent load factor line as indicated by the arrowed line in the occurrence diagram. It is noticeable that all loads up to 26 per cent K.W. occur at 100 per cent load factor and fall on the straight line in the occurrence diagram.

The loading possibilities of a generating station may be readily studied using the occurrence curve just described. Figure 7

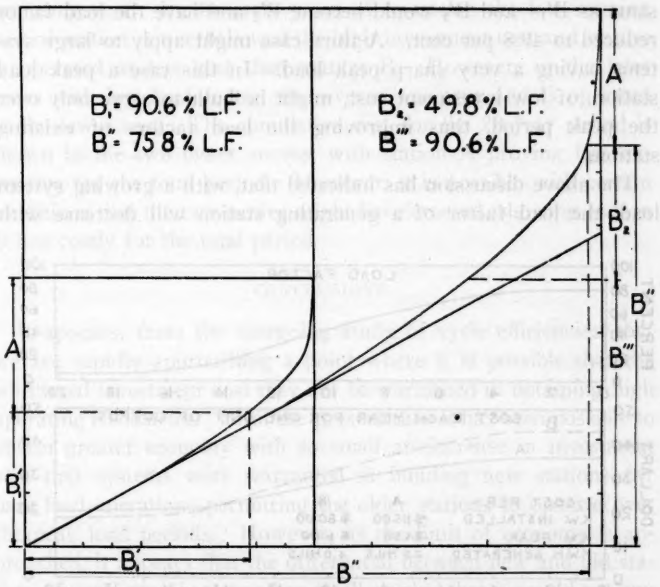



FIGURE 7.

shows two occurrence curves drawn on the same 100 per cent load factor line. The smaller or left-hand curve is just one-half as large as the larger one. For the smaller curve, assume a system with two generating stations, A and $B'_1$, each capable of supplying half the load, but $B'_1$ being the more efficient. In this case $B'_1$ would run as a base load station with a load factor of 90.6 per cent as indicated and A would operate as a peak load station. Now

let us suppose the system load doubled, as in the larger curve. Station A remains the same and continues as a peak load station, but now only supplies one-quarter of the K.W. load. The remaining load could be supplied in several ways. First, by enlarging $B'_1$ with the same type of apparatus; in which case it would become B″ and have the load factor reduced to 75.8 per cent. Secondly, and more likely, $B'_1$ would remain the same and a more efficient station, B‴, be built to carry the base load. In this case the new station would have a load factor of 90.6 per cent—the same as $B'_1$—and $B'_1$ would become $B'_2$ and have the load factor reduced to 49.8 per cent. A third case might apply to large systems having a very sharp peak load. In this case a peak load station, of low investment cost, might be built to serve only over the peak period, thus improving the load factors of existing stations.

The above discussion has indicated that, with a growing system load, the load factor of a generating station will decrease with

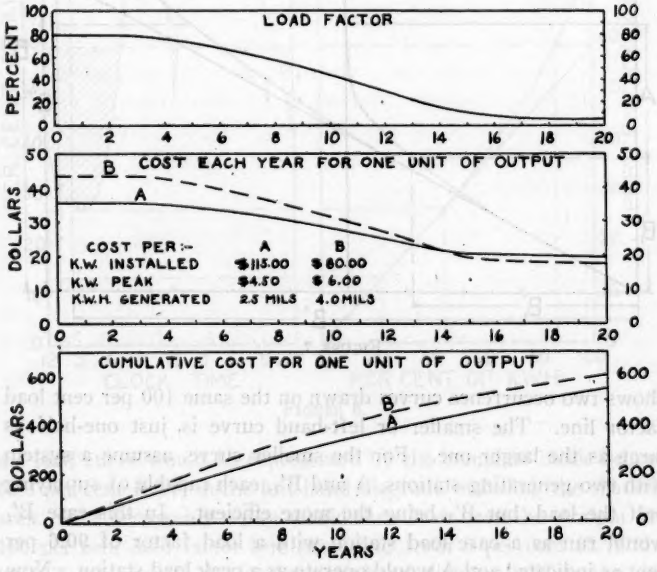



FIGURE 8.

years of service due to the construction of new, more efficient stations. This fact materially effects the balance of investment and operating cost in choosing a new type of installation.

The upper curve of Figure 8 is an estimate of the reduction of load factor on a generating station over a 20-year period. The curves in the middle block show the cost each year for one unit of output for two stations operated at the estimated load factors. Station A is a high investment cost station, with low operating cost, and station B is a low investment cost station, with a high operating cost. In the early years of operation the low running cost of station A overbalanced the high investment cost and station A operated at less cost than station B. However, in the later years with low load factor the reverse was true and B operated at the lower cost. The cumulative cost for the 20-year period is shown in the two lower curves, with station A proving less expensive for the total period. However, it is possible with certain conditions that the two curves might have crossed, showing station B less costly for the total period.

## CONCLUSIONS.

It appears, from the foregoing study of cycle efficiencies, that we are rapidly approaching a point where it is possible that the additional investment cost may not be warranted in obtaining high operating efficiencies. Until the present time it has been possible to obtain greater economy with so small an increase in investment cost that systems were warranted in building new stations for base load operation; permitting the older stations to operate over the peak load periods. However, as the limit of economy is approached, it appears that the differential between new and old station economy may become of so small a value that it would not pay to drop load from the old stations in order to carry base load on the new. In this case the building of peak load stations, with low investment cost, might be warranted at intervals as an alternate to the present policy of base load installations.

It is granted that some of the factors mentioned herein are not easy to evaluate, but the time has arrived when they must be carefully considered in all studies relative to new installations if the best balance of economy is to be maintained.

# DISCUSSION OF THE VARIOUS PULVERIZED FUEL SYSTEMS.*

By Henry Kreisinger, Research Engineer, Combustion Engineering Corps.

The use of pulverized coal as fuel for making steam has been developed along two lines, namely:

1. The storage system.
2. The direct firing or so-called unit system.

In the storage system the process of preparation of pulverized coal is independent of the process of burning coal. The pulverized coal is first prepared and delivered to the storage bin located at the place of its use, and is then burned as load conditions require.

In the direct firing system the process of preparation of pulverized coal and the process of burning it are dependent on each other. The coal must be pulverized as fast as it is burned, and in turn it must be burned as fast as it is pulverized.

Each of these systems has its advantages and disadvantages. Each system also has its adherents who try to promote and improve it by eliminating its natural weaknesses. These efforts have been going on during the last few years with various degrees of success for both systems.

The adherents of the direct firing point out that the storage system is too expensive and too complicated because dryers, pulverized coal transports, pulverized coal bins and feeders must be bought, operated and maintained. Also that in the operation of dryers, mills and air transports of the storage system, the dust elimination presents a difficult problem for solution. They concede that the storage system is more reliable, more flexible and provides better condition for efficient combustion, but that these advantages are not of sufficient magnitude to offset its disadvantages.

* Paper read before a joint meeting of the Washington Sections of the A. I. E. E., A. S. M. E., and A. S. N. E.

The engineer who believes in the storage system feels that the objections to it are not entirely inherent and that they can be to a large degree eliminated. He, therefore, is modifying the designs of the storage system in an effort to simplfy it and to reduce its cost, and at the same time to maintain its advantages.

The storage system plants built or designed within the last few years varied greatly in the amount of apparatus involved, and in the location and the arrangements of the different parts of the apparatus. The design of these plants varied from separate coal preparation building located some distance from the boiler plant to designs in which the coal preparation equipment was placed right in the boiler room. Between these two extremes various combinations were made, depending whether the esthetic or the economic reasons predominated. The taste and the prejudices of the designer and the future operator also had a great deal to do with the design of the plant.

There are numbers of installations showing the different arrangements in the design of the plant and location of the coal preparation equipment.

Among the plants with the separate building for the coal preparation equipment are: The Lakeside Station of the Milwaukee Electric Railway and Light Company, the Fordson plant of the Ford Motor Company, the Trenton Channel Plant of the Detroit Edison Company, the Holtwood Station of the Pennsylvania Water and Power Company, the Gould Street Station of the Consolidated Gas, Electric Light and Power Company, the plant of the Narragansett Electric Lighting Company of Providence, and the Middletown Station of the Metropolitan Edison Company.

Among the plants with the coal preparation equipment in the boiler room are: The Peoria Plant of the Illinois Electric Light and Power Company, the Manchester Street Station of the United Electric Railways Company of Providence, the Walkerville Plant of the Ford Motor Company, Walkerville, Canada; the Toronto Station of the Penn Ohio Power Company. The St. Paul Plant of the Ford Motor Company goes even as far as to put coal preparation equipment in the same room with the boilers and turbines.

The Lakeshore Station of the Cleveland Electric Illuminating Company has its coal preparation equipment placed in a separate room, but this room is located alongside of the boiler room.

The kind, the amount and the arrangement of coal preparation equipment varies with its location. Plants with separate coal preparation plant usually have air transports for moving the pulverized coal from the preparation plant to the feeder bins in the boiler room. Other coal transports in such designs are considered usually impractical, although the Fordson Plant has been operating a screw conveyor about 600 feet long for a number of years with entire satisfaction.

The use of air transport requires much more additional apparatus than the use of screw conveyors. For example, there must be a complete air compressing plant, including compressed air storage tanks. The coal must be dried to a very low percentage of moisture in order that it may be transported successfully with an air transport. This drying requires large dryers with fans and cyclone separators for partly cleaning the air carrying the moisture out of the dryers. Even with the cyclone separators such dryers may discharge an objectionable quantity of dust into the atmosphere.

During the transportation with air transports, the coal must be mixed with certain quantity of air which must be vented from the storage bin where the coal is delivered. It always carries an objectionable quantity of dust along with it. Thus, the air transport greatly aggravates and enlarges the dust nuisance around a pulverized coal plant.

On the whole, air transports are bad because they make bad arrangements of the coal preparation plant possible.

The main reason for placing the preparation plant away from the boiler room and turbine room is the cleanliness. Even though the preparation plant may be kept clean, a large amount of dust is always raised by the unloading of the raw coal at the plant. The logical place for the unloading plant is close to the coal preparation plant. If the coal preparation plant is placed near the boiler room and the turbine room, the dust raised by the unloading of the raw coal is carried to the boiler room and to the turbine room and makes it dusty and dirty.

Most of the large central stations have coal storage near the plant on which they can draw for a coal supply in case of irregular coal delivery. Placing coal into the coal storage or taking it out is a dusty operation and spreads coal dust over the neighborhood.

For this reason the coal storage is usually some distance from the main plant, and inasmuch as the handling of coal from the coal storage into the preparation plant raises clouds of dust some designers prefer to place the preparation plant near the coal storage away from the main plant in order to keep the dust from the boiler room and the turbine room.

The regulations of the National Board of Fire Underwriters prescribed that the preparation equipment should be in a separate building or at least in a separate room. The supposed reason for such rules is to confine fires and explosions to the preparation plant and keep such fires and explosions as far from the main plant as possible.

In the days when the first pulverized coal plants were decided upon and designed, there was a great deal of agitation against the pulverized coal by stoker manufacturers and the danger of explosion and fires was much talked about. Comparatively little was known about the handling of pulverized coal and its behavior under various conditions. Those who knew the least about it talked the most. The designer of the plant therefore took benefit of the doubt and placed the preparation plant some distance from the boiler room and the turbine room. At that time move was also started to make rules for the safe installation and operation of pulverized coal plants, and the design of separate preparation plants was at once adopted as the safest design. Little thought was given how such rule would handicap the development of the use of this new form of fuel.

During the last few years engineers have learned more about pulverized coal and the ways to handle it. They know under what conditions it is dangerous and how to avoid such conditions. They also realized that the old design, with the preparation plant far from the boiler room, was too expensive and that it presented complicated dust problems. Instead of assuming that the preparation of powdered coal was a dusty process anyhow and that the best way to solve the dust problem is to remove it far from the main plant, the designers have now realized that a better way to handle the problem is to design the preparation plant so that it can be kept clean, and place it near the boiler room, if not in the boiler room.

We can, therefore, say that there is at present a decided effort to put the preparation equipment near or even in the boiler room, even though the Underwriters' rules make them to place fire brick partition walls around them. The economic reasons begin to dominate over false esthetic reasons and superstition.

By placing the preparation equipment near or in the boiler room, pulverized coal conveying apparatus is greatly reduced and in the majority of cases dryers can be entirely eliminated. What drying of coal needs to be done can easily be accomplished in the mills by supplying the mills with heated air and drying the coal during the process of pulverization.

By drying coal in the mill moisture up to 5 per cent can be easily removed. In extreme cases as much as 8 per cent can be removed from the coal if necessary. However, such cases are indeed very few.

Pulverized coal comes out of the mill system at much lower temperature than is the case when coal is dried in separate dryers and delivered hot to the mill. With coal drying in mill the temperature of the pulverized coal seldom exceeds 125 degrees Fahrenheit.

The heated air needed for mill drying can be taken either from the main air heater on the boilers or atmospheric air can be heated by a separate steam air heater. In either case the installation is much simpler and cost less than separate coal dryers. The heated air admitted into the mill for drying purposes must be taken out of the mill system. If the preparation equipment is in or near the boiler room this air vented from the mill system can be used as primary air and thus any dust that may be carried out of the mill with this air is readily disposed of, without the dust becoming a nuisance. If the preparation plant were some distance from the boiler room and the air vented from the mills could not be used as primary air, it would have to be vented through air washers and the dust removed from it before discharging the air into the atmosphere.

It has been said in the past that dryers were a necessary nuisance. By placing the preparation equipment near or in the boiler room and doing the necessary drying in the mill, separate coal dryers are becoming just plain nuisances.

The adherents of the unit system frequently point to the pulverized coal bins of the storage system as an unnecessary additional expense. This objection may be valid when the preparation plant was some distance from the boiler room and the pulverized coal had to be conveyed by air coal transports. In order to be insured against possible shortage of pulverized coal in the boiler room by the breakage of the air transport, the pulverized coal bins were made large to have "storage" of pulverized coal.

When the coal preparation equipment is placed near or in the boiler room, this necessity for large pulverized coal bin is eliminated, because there is no danger of running short of pulverized coal on account of breakage of conveyors. The pulverized coal bins can be made small. In fact, the combined capacity of the pulverized coal and the raw coal bins does not need to be greater than the storage bin for the raw coal of the direct firing system.

In short, the progress in the simplification of the storage system and in the reduction of its cost is manifested in the tendency to place the coal preparation equipment near or in the boiler room. Such designs eliminate separate dryers and reduce the powdered coal conveying apparatus to minimum. Such designs also eliminate entirely the dust nuisance. With such designs the cost of the storage plant is brought down close to the cost of the direct firing system, and in some cases even below it.

The appeal of the direct fired system of pulverized coal has been always strong to those in search of simplicity and low first cost. What could be simpler than a pulverizer placed in front of the boiler under a raw coal bunker, substantially as used for the stoker fired job? The mill draws the coal from the raw coal bunker, pulverizes it and discharges it directly into the furnace. No dryers, pulverized coal transports, bins and feeders with which to contend.

Such is the story of those who favor this system and those who for various reasons depend for their daily bread upon the use of the direct firing system in competition with the storage system. One naturally asks the question, if the problem were so simple as it at first appears, why has the storage system made progress at all against the apparently more simple direct fired system? A little investigation discloses the fact that although the direct fired system was old before the storage system was invented, the large ma-

jority of noteworthy installations use the storage system. There is apparently more to this question than appears on the surface.

The engineer who has decided in favor of the storage system, perhaps after thorough investigation of both systems, may tell us the following reasons why he has made such decision.

In the direct firing system the mill is the all-important factor. If its performance is faulty in any way, the entire installation suffers by its defects. With few exceptions the mill used for direct firing is of the high speed beater type, of which there are many makes on the market, but all have the same fundamental characteristics and weaknesses.

1. They all have definite maximum capacity, which, if exceeded, the mill chokes without warning.

2. Maximum capacity depends on the moisture of coal. In rainy weather coal becomes wet and capacity is greatly reduced. Hot air must be supplied to mill for drying coal and to make possible pulverization of wet coal with any satisfactory results. Wet coal does not make coal dust cloud that ignites easily.

3. All have high wear of the pulverizing parts, and therefore, high maintenance.

4. Feed of raw coal to mill irregular because the size and moisture of coal varies. This variation in feed is particularly aggravated at low loads.

5. Feeding of coal to mill may be completely stopped by arching of coal in the raw coal bunker, or by pieces of wood, rock, iron or other foreign substance getting into the feeder and stopping it. In such cases the fire may go out, and may have to be relighted with a torch. Load may be lost.

6. Sudden break down of the mill without warning puts the entire boiler out of service, resulting in a loss of load.

7. Reduction of mill capacity and of fineness of coal with wear on the mill. In order to make up this defect the mill must be oversized.

8. The maximum mill capacity and power consumption takes place simultaneously with the peak load on the station.

9. Inability to carry wide range of ratings unless more than one mill is used per boiler.

10. Difficulty to divide coal from large mill to many burners.

Notwithstanding the above weaknesses, the direct firing system has its legitimate uses. There are many industrial plants which may not be able justly to stand the higher first cost for the storage system, or which have not the room necessary for the installation of the storage system, but wish to take advantage of pulverized coal.

Small old boilers are usually much more easily converted to pulverized coal by the direct fired system than they would be by the storage system.

Standby Central Stations and Central Heating Plants which have large seasonal load variation find the direct firing system more economic than the storage system.

Improvements in the direct firing system have been centered almost entirely on the improvement of the mills.

The mills have been made more reliable against breakage. Means for a better control of the fineness of the coal have been incorporated.

The feeding to the mill has been made more certain by providing small bins just above the mill from which coal is fed to the mill. Such mill bins are filled about once in an hour. Because of this frequent emptying and filling of the bin the coal arching over the feeders is almost entirely eliminated.

The mills are built in larger sizes so that fewer mills are required for large steam generating units.

More recently mills have been applied to direct firing with success which hitherto have not been thought adaptable to direct firing.

It is very doubtful that either system will ever entirely displace the other. Both will improve and each will find its use in plants where their natural advantages will show up the best.

# SEA-GOING PULVERIZED FUEL.*

By C. J. Jefferson, U. S. Merchant Fleet Corporation.

The preliminary test plant work, the actual installation and the plant performance of the first offshore pulverized fuel plant, the S. S. *Mercer,* has been described ably and concisely to this meeting by Commander J. S. Evans,† with whom I have had the distinct pleasure of carrying on this experiment—an experiment which has been carried on for the benefit of the American Merchant Marine by the Governmental departments, the U. S. Navy and the U. S. Shipping Board.

The S. S. *Mercer* was and is an experiment. It is not a final solution to the problem by any means, but it has and is serving as a seagoing laboratory wherein practical operating problems are being worked out, the solution of which will be of considerable value in developing an ultimate standard of equipment and scheme of operation.

I feel that it is most opportune at this meeting of the combined mechanical, electrical and naval engineering societies, to explain just why certain controversial items have been done as they have been done on the S. S. *Mercer.*

First, why was an oil burning vessel selected?

Because this experiment was to be carried on in a seagoing cargo vessel which has to meet traffic schedules, and while the preliminary test work at the Philadelphia Navy Yard gave promise of success, still there was no precedent from which it was possible to predict what would happen when the vessel was rolling and pitching and the equipment was subjected to the strains attending seagoing conditions.

Therefore it was considered advisable to have oil fuel in reserve in case of failure on the part of the pulverized fuel plant.

* Paper read before a joint meeting of the Washington Sections of the A. I. E. E., A. S. M. E., and A. S. N. E.

† Note. It is expected that a complete report of the tests conducted by the Fuel Oil Test Plant will be published in the August issue of this Journal, in a paper being prepared by Comdr. J. S. Evans.

Furthermore, prior to the installation on the S. S. *Mercer* it was not at all certain that the pulverized fuel plant could be regulated quickly enough to satisfactorily meet maneuvering conditions and it was believed that until further development work had been carried on that it would be necessary to maneuver with oil. I am glad to state, however, that such was not the case, as we found in service that the vessel maneuvered beautifully with pulverized fuel.

During the initial voyage of the S. S. *Mercer,* oil fuel was used on several occasions at sea, while making repairs to equipment which had been changed to meet the load of seagoing service. It would not have been necessary to resort to oil as a safety measure, but it did enable the vessel to meet her traffic schedule. The total amount of oil used on the New York to Rotterdam passage was 100 barrels and on the Rotterdam to New York passage 90 barrels.

This amount of fuel is remarkably small considering the experimental conditions that attended this voyage, and I would not be at all surprised if it would not be greatly exceeded on subsequent voyages when further experiments are being conducted, especially with various grades of fuel—because I feel that on this initial voyage we were especially fortunate in the absence of slag and ash in our furnaces and rear connections.

Lacking a satisfactory tight coal valve, it was necessary to use oil a great part of the time in port, as the minimum steam supply from three-furnace operation under reduced port load conditions such as prevail when not working cargo, was excessive and resulted in popping of safety valves or else a cut-in, cut-out type of operation that required considerable attention on the part of the operating personnel.

This condition was corrected in several instances by blanking off the coal line with a blind gasket. This, however, is only a makeshift and development work is now being carried on to produce a quick closing, tight and reliable coal valve.

A second point that has been brought up several times is why the " Unit" instead of the " Bin System" was adopted.

The answer is for safety reasons. With the unit system there is but a very small amount of pulverized coal in existence at any one time, and this is all moving with considerable velocity towards the furnace, with access to the atmosphere through the stack.

Coal stored in bins subjected to moisture of condensation, characteristic of seagoing vessels, will cake and cease to be pulverized fuel, but rather will become blocks of coal concrete, utterly unsuitable for combustion purposes.

When the unit system was decided upon, the question then arose as to what type of mill should be used.

Mr. Kreisinger in his discussion of the Unit versus the Bin System has practically explained the answer to that question, as the objections raised against the unit system apply almost exclusively to the high speed paddle type of mill.

The problem of properly balancing the rotor of a high speed mill arose during the work at the test plant, as did also the failure of this type of mill to produce the superfines necessary for combustion in the limited combustion space of marine boilers, and when this was coupled with the falling off in fineness of grind at increased capacity demand rate, as well as with varying grades of coal, it became quite evident which type of mill suited our particular problem best, and it was for these reasons that the slow speed tubular type of air swept mill was used.

The advisability of installation of a crusher plant has also been discussed. To obtain uniform quantity from a pulverizer, it is necessary that a uniform quantity be fed to it, and a feeder that will maintain a uniformity of feed which can be regulated by the operator, demands sized coal or at least coal which does not exceed a reasonable maximum size, which is approximately 1½-inch lumps.

If sized coal can be bunkered, the crusher plant is not necessary. But, as stated before, the S. S. *Mercer* is an experimental ship, and it is proposed to bunker her with various grades of coal, and such being the case, she must be prepared to bunker with run of mine as well as slack and the sizing of the bunkered fuel be accomplished on board. Therefore, in her case, at least, the crushers are justified and in fact for offshore work, where vessel will bunker in various ports, the crusher really becomes an essential part of the equipment.

The S. S. *Mercer* has successfully demonstrated the reliability and safety of pulverized fuel by sailing across the Atlantic during the winter months and meeting her traffic schedule. This she has

done with two boilers only in service, while her sister ships, burning oil or hand fired coal, have used three. She has indicated the economy possible with pulverized fuel, and as she sails on future voyages modifications will be made to her equipment, modifications which work at the test plant indicates will produce improved results, but which must be given sea trial before acceptance.

In the meantime, apparatus of competitive manufacture is being tested out at the Philadelphia Navy Yard Fuel Test Plant.

Pulverized coal has made its first cruise and has taken unto itself seagoing ways!

# REMARKS ON THE USE OF PULVERIZED FUEL FOR MARINE INSTALLATIONS.*

By Captian R. D. Gatewood (C.C.), U. S. Navy.

I wish to emphasize the point that both Commander Evans and Mr. Jefferson have stated: Namely, that we consider the S. S. *Mercer* to be the extension school of the Fuel Test Plant, where the experiments which have been started at the Test Plant can be carried out afloat.

On her we have demonstrated, as Mr. Kreisinger stated, that pulverized coal is a "good natured fuel," that it can be handled safely and give reliable service. The economies which may be expected from this type of fuel have been indicated, and the points of attack which must be made to produce a standardized equipment and a standard method of operation have been crystallized.

Relative to the safety feature, I feel that the bin system is not at all suitable for use on shipboard, and that it would be practically impossible to have such a system approved by insurance underwriters or the Steam Boat Inspection Service even if such a system were desirable.

Nor is it desirable as the addition of coal dyers, the introduction of the problem of moisture in the pulverized fuel due to sweating of the ships' sides, or tanks, the expense of securing absolutely tight tanks (and this last item is no small expense, either, as pulverized coal can leak out of joints that would be absolutely tight against either water or oil), all react unfavorably towards a bin system installation.

I feel that the problem of pulverized fuel for marine use definitely resolves itself into a unit mill solution.

As to the type of unit mill. The objection to the use of high speed mills, as cited by Mr. Kreisinger, explains why we adopted the slow speed mill. I do feel, however, that considerable develop-

* Paper read before a joint meeting of the Washington Sections of the A. I. E. E., A. S. M. E., and A. S. N. E.

ment work should be given to mill design by reduction engineers so as to produce a mill of considerably less weight and size for marine use. By this, I mean a mill which will consistently give a large percentage of superfines with various grades of coal over a fairly wide range of capacity, and at the same time be of such a size that it can be installed in the fireroom itself without making structural changes in the vessel to accommodate it.

The installation of conveyors and crushers on the S. S. *Mercer* is necessary in order that she can serve her purpose as an experimental laboratory and permit bunkering with various grades of coal, both at home and abroad.

However, I feel that the ultimate design will eliminate these two items, as the bunkering of pulverized fuel ships will be made with slack coal, as this type of combustion lends itself very readily to the use of the grades of fuel which are not at all suitable for combustion in any of the other methods of firing.

I recently heard a speaker state that in life, when a thing is accomplished, it ceases to be of importance other than the aid which it can give to future progress. So it is with the problem of pulverized fuel for marine purposes. The successful voyage that has been made by the S. S. *Mercer* is an accomplished fact which has definitely established certain engineering features, but of greater importance, it has pointed out the line of attack which we must carry on.

# COMMENTS ON THE USE OF HIGH PRESSURES.*

By A. R. Smith, Engineer, General Electric Company.

That the use of high pressure steam for the generation of electric power is feasible is not questioned in stationary power plant circles; that it is economically sound is quite generally accepted and that it has been found commercially successful cannot be denied in view of the increasing use of higher steam pressures in public utility plants, where reliability is seldom sacrificed for economy and where the return on every dollar invested is pretty accurately predetermined.

High steam pressures may be grouped in three general classes, 400, 600 and 1200 pounds. The 600 and 1200 pounds pressures are, however, more generally employed where the fuel cost is high, or the load factor good. Pressures below 350 pounds cannot usually be justified even with low cost fuel and moderate load factors, because the investment in coal handling equipment, condensers, circulating water supply and other accessories is sufficiently reduced as the result of the improved turbine water rate to offset the additional cost for high pressure boilers.

Probably the chief reason why high pressure steam has not been more generally employed in marine work is the possibility of the contamination of the feed water as a result of condenser leakage. Even with 200 pounds steam pressure, the modern highly perfected boiler should not be expected to operate with salted water, nor the modern high efficient steam turbine to operate with slugs of water or with encrusted or corroded buckets. This is obviously more imperative as the steam pressure is increased, but there seems to be no reason why a determined effort should not be made to minimize or prevent the contamination of the feed water which would then permit the use of a more efficient heat cycle.

Certainly there is no need of tolerating tube gland leakage when rolled joints are so satisfactory and expansion can be provided for by means of bent tubes or floating tube sheets. Tube corro-

* Paper read before a joint meeting of the Washington Sections of the A. I. E. E., A. S. M. E., and A. S. N. E.

sion from salt water cannot be stopped or materially delayed with the present knowledge on the subject, but the contamination of the condensate can be minimized by dividing the water boxes in two or several sections so that any section developing a leak may be quickly shut off. Salted condensate can be detected and an automatic alarm sounded by means of an electric water conductivity meter.

There is a means of preventing the condensate from becoming salted, which consists of surrounding each condenser tube with a second tube and circulating condensate through the small annular space between the two tubes at a pressure just a trifle higher than the salt water pressure. With only a slight difference in pressure much less water will flow through a fracture and the condensate will flow into the salt water instead of the salt water flowing into the condensate.

Such a duplex condenser would take but a little more space than the usual form of condenser, and it will introduce an additional heat drop which must impair the vacuum, at least during the warm water season, when the vacuum maintained will be a function of the water temperature rather than a question of air removal capacity. On the other hand, this loss of heat head resulting from a slightly poorer vacuum is much less than the heat head to be gained by resorting to higher steam pressures.

A secondary reason frequently given for not using high pressure steam in marine practice is that it is not suitable for the reciprocating steam driven auxiliaries. From the standpoint of simplicity, maintenance and efficiency, electric driven auxiliaries can invariably be justified, but assuming that there are some instances where steam driven auxiliaries are desirable, then 150 pounds steam with little or no superheat, as may be desired, can be produced in a steam regenerator using high pressure superheated steam and the condensate of this low pressure steam used for makeup for the high pressure boilers. In brief, this is simply an evaporator for supplying the makeup and incidentally transforming the pressure and temperature of such steam as must be used for auxiliaries.

It is quite safe to say that feed water for 200 pound plants requires more attention and is being given more attention than in

the past; that 400 pounds is practical even with reasonable precaution in maintaining pure feed water; and that with means of preventing any possible salting of the feed, 600 or 1200 pounds may shortly find many advocates. In brief, the seat of the trouble is the possibility of condenser leakage, and the sooner that this possibility is minimized or prevented, the sooner will the naval architect be in a position to take advantage of available and suitable economic methods, which have become more or less standard practice in stationary plants.

The designer also has the choice of selecting higher superheat without resorting to extreme pressures. Even with bad feed water, the convection superheater has not been troublesome, and such steam as may be needed for auxiliaries can be desuperheated in several ways; consequently, the foregoing arguments do not prevent the use of highly superheated steam.

With 200 pounds steam pressure, the superheat can be greatly increased over the prevailing superheat temperatures without any of the superheat entering the condenser.

Stationary plant practice now dictates 750 degrees F. as the practical uppermost limit, with most modern plants operating at from 700 degrees to 725 degrees F., but this is only a temporary standard or limit and the chances are that by the time that some of the ships now being planned are on the high seas a few years, much higher temperatures will be considered practical and possibly standard for stationary plants.

The policy should be to produce steam at as high a pressure and temperature as is considered practical and deliver all such steam directly to the turbine which can be constructed for pressures or temperatures that are safe for the other apparatus. The water of condensation should be returned as promptly and directly as possible to the boiler. More importance should be attached to the steaming fluid. It is not the common water used for general service, although it is obtained from that commodity. It should be conserved, tested and protected against contamination and regarded as a steaming fluid entirely apart from other waters in power plant service.

# MEMORANDUM CONCERNING HIGH PRESSURE STEAM INSTALLATIONS FOR MARINE PROPULSION.*

By Charles P. Wetherbee.

Steam holds the undisputed field for propulsion of all ships in which the horsepower per shaft exceeds 3500. Installations exist where this power rises to 45,000 per shaft.

In the lower part of this steam field it is mechanically possible to use reciprocating engines, but above 10,000 horsepower per shaft the steam turbine is the only source of power available today.

A little more than a year ago the Parsons Marine Steam Turbine Co. demonstrated on the *King George V* that it is entirely feasible to use steam at a boiler pressure of 550 pounds per square inch and heated to 750 degrees Fahrenheit.

The Canadian Pacific Company has recently completed some new steamers with 350 pounds boiler pressure and a high superheat. I have not yet seen any report of results obtained on these ships.

In the United States, shipowners have not yet been able to see where they are financially justified in going to high pressures and temperatures. They are unwilling to take the risk of being pioneers in any innovation unless the operating saving involved will authorize the extra cost of the installation within six or seven years. So far our designing marine engineers have not been able to convince them on this score.

There are several reasons for this.

Unlike an electric power house, the main propelling units of a ship are in use only from 50 to 60 per cent of its commissioned life, even under the best of management.

There are many steam consuming auxiliaries in a ship that are necessary and in practically continuous operation at sea and in port twenty-four hours per day. They have nothing to do with

* Paper read before a joint meeting of the Washington Sections of the A. I. E. E., A. S. M. E., and A. S. N. E.

the main propelling units. Most of them are pumps of moderate capacity. Many of these have to be able to suck and frequently they must discharge at high pressure. No satisfactory unit has yet been devised for this service except the direct acting reciprocating steam pump and this cannot use high temperature steam. The ship contains other auxiliaries of the rotary type, such as electric generators, forced draft blowers, condenser circulating pumps and boiler feed pumps, that can use high pressure and temperature steam if the necessary money is spent on their construction, but the saving in these smaller units is not as great proportionately as in the large propelling turbines.

The thermodynamic gain by using high pressure and temperature steam for driving the auxiliaries is, therefore, small, and what economy is obtained in such a ship must come principally from the main propelling units which are only in operation part of the time.

The saving in the monthly fuel bill on a ship cannot, therefore, be as great as that in a power house operating under the same high pressure steam and temperature conditions.

Then there are the matters of weight and space. Air heaters, or feed water heaters, or both, in the boiler uptakes are vitally necessary with high temperature steam, because, if the additional heat in the uptake gases of such boilers is not recaptured, the gain by using a wider range in the working temperature cycle of the turbine is largely lost through the smoke pipe. On a merchant ship, it is usually possible to find space for these air heaters and the weight is not a very serious consideration, but it is, nevertheless, a direct charge to the high steam plant in any comparison with one using lower pressures.

Much greater care in piping, pipe fittings, valves and lagging must be used. All non-conducting covering must be much thicker, and such covering must be applied to surfaces usually left bare on the ordinary job.

All of these things cost money and the interest, insurance and depreciation on this extra investment goes on continuously throughout the life of the ship in spite of the fact that the apparatus represented by it is used not over 60 per cent of the time.

These are some of the difficulties that the designer is at present trying to reduce or eliminate in his endeavor to make the very

real thermodynamic economy attending the use of high steam pressures and superheat financially attractive to the shipowner.

In warships the question of extra first cost is not so important. The advantage of increased fuel economy reflected in greater cruising radius on a given fuel capacity has to be balanced against the greater weight, vulnerability and space occupied.

I have not yet seen a satisfactory way to install the boiler air heaters on a warship.

The main propelling machinery is in use even a smaller fraction of the total time than on a merchant ship, and a heavy battery of auxiliaries is in continuous operation.

There might be advantage in developing an arrangement where high steam pressure, with its accompanying increase in cruising radius, could be used at the normal cruising speed without attempting to provide for it at higher powers which are seldom used and where cruising radius is not so important.

# MARINE BOILERS FOR HIGH PRESSURES.*

By J. B. Stillman, of Babcock & Wilcox Co.

The increase in use of higher steam pressures and temperatures ashore clearly indicate the practical and economical value of such installations. At the present time there are more than 5,000,000 square feet of heating surface of boilers manufactured by the Babcock & Wilcox Company alone using pressures of from 350 to 450 pounds and more than 1,250,000 square feet in boilers using pressures of from 650 to 1400 pounds in the stationary field in the United States.

A number of years ago the marine practice was appreciably ahead of the stationary practice in the matter of high pressures, 250 to 300 pounds in the marine field being relatively common when the stationary was using 200 to 250 pounds as a standard. The first 500-pound pressure boiler of which the writer has knowledge, built by the Babcock & Wilcox Company, was manufactured in 1905 and installed at the Annapolis Experiment Station of the United States Navy for testing safety valves. This boiler was of the Babcock & Wilcox Marine Type. At this time pressures of 295 to 300 pounds were relatively common in U. S. Navy work.

It was not until 1915, when the Public Service Corporation of Northern Illinois made an installation of boilers suitable for 350 pounds pressure, that the stationary practice began to step ahead of the marine. But from that time the gap has been growing greater and greater.

In 1922 at Philo, Ohio, the American Gas and Electric Company built a plant suitable for 650 pounds pressure, but operated at 550 pounds, and 700 degrees temperature at the turbines, where for the week ending April 18, 1925, they secured a kilowatt for 13,715 B.T.U. per hour. As they use coal having a B.T.U. value

* Paper read before a joint meeting of the Washington Sections of the A. I. E. E., A. S. M. E., and A. S. N. E.

of 10,390 per pound as fired, they are in regular operation in this plant securing a kilowatt for 1.32 pounds of coal per hour. Assuming the use of 13,000 B.T.U. coal, which is more common for marine practice, they would have secured a kilowatt for 1.05 pounds per hour. This is equivalent to .78 pound of coal per S.H.P., or assuming the use of fuel oil containing 18,500 B.T.U. per pound a S.H.P. per .55 pound of oil per hour. The results which have been secured at the Miami Fort Station of the Columbia Power Company near Cincinnati, Ohio, where the steam is generated under somewhat similar conditions of pressure and temperature as at Philo, using pulverized coal fuel, show similar efficiencies in terms of B.T.U. per kilowatt output. These two stations indicate in a general way what may be anticipated with relatively high pressures and temperatures in properly designed steam power plants.

It is of interest to note that in going to the higher pressures in stationary practice the cross drum type of Babcock & Wilcox boiler, which has been standard with us for years in the marine field, is the accepted design for this purpose, it being more suitable for these pressures than the longitudinal type of Babcock & Wilcox boiler that reigned supreme in the stationary field during the 200 to 250-pound pressure era. Accordingly, it will be readily appreciated that the Babcock & Wilcox marine boiler is ideally suited for these higher pressures, the only change necessary being an increase in the thickness of the drum sheet to meet the pressure specified.

During the last year two ships have gone in commission with pressures exceeding 300 pounds, these being the S. S. *Dixie,* of the Southern Pacific Company, which operates at 350 pounds pressure and 200 degrees superheat, oil fuel, and the S. S. *Carl D. Bradley,* of the Bradley Transportation Company, which uses steam at 325 pounds pressure, 290 degrees superheat, using coal fuel with underfeed stokers. We shall refer to the economies of these ships later on.

At this point it might be well to indicate why it is desirable to operate at higher pressures, as well as temperatures. Frequently the point has been raised that if the total B.T.U. in a pound of low pressure steam is as high as the total B.T.U. in a pound of

higher pressure steam, the same economies should be secured, provided both turbines are exhausting to the same vacuum. This is not the case, as shown by the following tabulation, which assumes the use of a 100 per cent efficient turbine in extracting the available heat in each case. The back pressure is assumed at 2 inches mercury:

| | | | |
|---|---|---|---|
| Steam Pressure (absolute), pounds | 600 | 400 | 220 |
| Steam Temperature, degrees | 700 | 677 | 660 |
| Superheat, degrees | 213.5 | 232.2 | 270 |
| Total Heat, 1 pound Steam (B. T. U.) | 1,350 | 1,350 | 1,350 |
| Heat in Steam at 2 inches (B. T. U.) | 885 | 910 | 945 |
| Heat given up to turbine (B. T. U.) | 465 | 440 | 405 |
| Per cent moisture in the steam entering condenser, per cent | 21 | 18.7 | 15.4 |

From the above it may be noted that the 600-pound installation delivered 60 more B.T.U. per pound of steam to the turbines than the 220-pound job, this being an increase of nearly 15 per cent in the heat absorbed per pound of steam by the 600-pound installation as compared to that absorbed by the 220-pound installation, both starting with steam having a total heat content of 1350 B.T.U.

Actually due to friction losses in the turbines, etc., which are fairly uniform for all three pressures, the figures above are not secured in service, although relatively the difference between them remains fairly constant. As an illustration of this difference, instead of securing 21 per cent moisture in the steam going to the condenser from a 600-pound 700-degree installation, this will be nearer 11½ per cent, and in each case the theoretical percentage of moisture is approximately cut in half due to this friction. In a way this is fortunate, as it is not usually customary to try to operate turbines with much in excess of 15 per cent moisture in the steam to the condenser. If this condition exists it is usually desirable to introduce a reheat cycle somewhere in the system before the condenser is reached.

From the above it will be seen that it is not likely that pressures greatly in excess of 600 pounds will be used in the marine field for some little time, as to do so would require reheating, which introduces considerable complications in the piping from a marine point

of view, although such practice is universally common in the stationary field, where pressures in excess of 600 pounds are in service.

Riveted drums 42 inches in diameter may be made for pressures up to 750 pounds. Beyond this point it would probably be desirable to use forged drums as is customary in our stationary practice for these higher pressures. For marine work the riveted drum has the advantage of lighter weight and lower cost within the pressure limits to which it may be built.

At the present time the feeling is more or less universal in the stationary field to limit the total temperature of steam to approximately 750 degrees Fahrenheit. So far in marine practice we have not made any installations over 720 degrees. For marine work it will probably always be desirable to figure on a slightly lower total steam temperature than in stationary plants because of the fluctuation of the superheat that is liable to occur due to maneuvering, a condition which does not appear in the stationary picture.

With a Babcock & Wilcox type of boiler using a superheater located at the top of the first and second passes of the same, the amount of superheat which can be obtained is somewhat limited, because of the relatively low gas temperature available at the superheater. To meet the modern requirements of higher temperatures, we have gone to the interdeck construction, where the superheater is located part way down the first pass of the boiler. In this way it is possible to secure gas temperatures surrounding the superheater tubes which give the high superheat desired with a relatively small superheater surface, and, at the same time, sufficiently protect the superheater with from five to seven boiler tubes between it and the furnace, so as to prevent any danger of burning it in service.

Several years ago we conducted experiments at our Bayonne plant with a Babcock & Wilcox marine boiler using radiant type superheaters. The conclusions we arrived at based on these experiments were that if the superheater was sufficiently exposed to the furnace to give the high superheat desired it was very liable to burn out under maneuvering conditions. If we protected it enough by means of refractories or screening to insure a normal operating

life under maneuvering conditions, it was impossible to secure the high superheat desired in the small furnaces available in marine practice. To date we have never had occasion to change the above conclusions, and accordingly for the higher superheats we recommend the interdeck type of superheater for marine work.

Of necessity, as boiler pressures are increased, the temperature of the tubes rises accordingly, and the exit gas temperatures correspondingly increase, reducing the boiler efficiencies for a given amount of fuel burned. To overcome this condition in practically all modern high pressure stationary installations, either economizers or air heaters have been fitted to recover the heat in these higher temperature gases. As the pressures are increased in the marine field, we find a similar effort being made, as, for example, the S. S. *Dixie* and S. S. *Carl D. Bradley,* where air heaters are utilized.

Although our Stationary Department manufacture and install large numbers of economizers, we are not inclined to favor them for marine work as much as the air heaters. Because of the high pressures at which modern installations are operating, it is necessary to use steel tube economizers and these are subject to very rapid corrosion unless the air in the feed water is reduced to about one-tenth of a c.c. per gallon. This condition is practically impossible to secure unless a deaeration system is installed in the ship, as is done in a great many central stations. Because of the weight and bulk of these deaerators they cannot be considered seriously for marine jobs.

Consideration has been given to increasing the velocity of the water in economizers for marine practice to a point where the air in the water would not have an opportunity to attack the steel. Theoretically there may be something in this idea, but, unfortunately, in marine work the boilers are operating under port conditions so much of the time that the velocity of the water cannot be depended upon to prevent corrosion 100 per cent of the time. Also the economizer introduces a greatly increased number of high pressure parts and joints, which require attention, in a portion of the ship least accessible for convenient overhauling. An air heater, on the contrary, does not of necessity have to be absolutely tight, even when first built, and if some leaks do develop they do no

special harm, except to reduce the amount of air reaching the burners as compared to that entering the heaters. The weight of an economizer and of an air heater, to give the same reduction in uptake gas temperatures, is about the same.

One of the principal reasons, however, why we advocate air heaters for marine work in preference to economizers is the fact that the preheated air secured with an air heater is of considerable importance in improving combustion conditions obtaining in small marine furnaces. The smaller the furnace, the more important it is that the combustion be secured rapidly, and this the preheated air assists in doing very effectively, regardless of whether the fuel is oil or coal.

In our practice, we have developed a special air heater having horizontal tubes with the air passing through the center of same, the gases around the outside, these making three passes over the tubes. In service these air heaters have proved very satisfactory and efficient, and lend themselves readily to a compact duct arrangement at the rear of the boiler for conveying the heated air to the fuel.

Although it has been customary with Babcock & Wilcox marine boilers to use water cooled side walls in hand fired coal installations, for a great many years we have not found it commercially necessary or desirable to use them with oil fuel, pulverized coal or stoker firing. They increase the weight, introduce a large number of pressure parts in relatively inaccessible positions and so far have not been found necessary. With the high grade of firebrick available in the market at the present time, there is less and less need of considering water cooling in the small furnaces used in marine practice.

In naval work as distinct from merchant marine practice, the matter of weights is of *primary* consideration, and this has led to the almost universal adoption during the last few years of the three drum type of Express boiler using small tubes. The Babcock & Wilcox Express boiler, formerly known as the White Forster, is a representative illustration of this type where weights of approximately 12 pounds per square foot of heating surface have been secured for boiler, superheater, furnace lining and water. This is about half the weight per square foot of the average Babcock & Wilcox type of boiler as installed in merchant marine service.

Nobody has been more alive to the desirable features of high pressure and temperature in steam than the engineers of the Navy Department, but they have been handicapped in attempting to utilize these higher pressures and temperatures because of the inherent limitations of the three drum type of boiler when it is built for these pressures and temperatures. As the pressures are increased for this type of boiler, it is usually necessary to increase the thickness of drums or appreciably decrease the diameter of same. The latter is undesirable, especially in the case of the steam drums. Also, it is frequently necessary to increase the tube spacing, this appreciably reducing the efficiency of the boiler and also increasing the weight of the unit as a whole. The result has been that when studies have been made by the Navy Department to use increased pressures and temperatures, they have found that they have lost nearly as much in boiler room efficiency as they picked up at the turbines.

A short time ago we made a study for the Bureau of Engineering using an air heater on top of our Babcock & Wilcox Express boiler and found that with the higher pressures desired, it was necessary to add a 35 per cent air heater to give the same efficiency to the unit that they were securing with the same type of boiler at 300 pounds pressure. The air heater with the necessary ducts to the burners increased the weight of the unit three pounds per square foot of boiler surface, which eliminated the study from consideration in the naval ship proposed.

During the last year we have been working on this problem and believe we have now reached a point where we will be in a position to offer a boiler for Express service which at 600 pounds pressure will have the same or slightly better efficiency than our present Babcock & Wilcox Express boiler at 300 pounds. This new boiler will be as light or lighter than the present Express boiler per square foot of heating surface, and in view of the fact that less surface will be required for a given power in a high pressure ship, an appreciable saving in weight in the fireroom will be secured. There will also be a decided saving in space occupied in the fireroom by the boiler compared to the present three drum type of boiler for the same heating surface.

As an indication of the success that may be expected from using higher pressures and temperatures in marine work, we tabu-

| Name | Steam Pressure | Superheat | Shaft H.P. | Type of Firing | Efficiency Boiler and Super Heater | Efficiency Boiler S.H. and A.H. | Oil per S.H.P. all purposes | Coal per S.H.P. all purposes |
|---|---|---|---|---|---|---|---|---|
| | | | | | Percent | Percent | | |
| S. S. *Malolo* | 270 | 125 | 27200 | Oil | 80.0 | ... | .80 | ... |
| S. S. *California* | 275 | 110 | 17400 | Oil | 82 | ... | .802 | ... |
| S. S. *Bradley* | 325 | 290 | 4878 | Stokers | ... | 83.9 | ... | 1.05 |
| S. S. *Dixie* | 350 | 200 | 8000 | Oil | ... | 80.0 | .75 | ... |
| S. S. *Virginia* | 300 | 200 | 17000 | Oil | 80.5 | ... | .75 | ... |
| 6500 Ton Collier | 400 | 250 | 2900 | Pulv. Coal | ... | 82.5 | ... | 1.02 |
| 3300 S.H.P. Freighter | 400 | 230 | 3300 | Oil | ... | 83 0 | .63 | ... |

late a list of ships in which the steam conditions may be regarded as a step toward higher pressures and temperatures. The data given for the first three ships are based on actual performances obtained from the ships in service. The other studies are for installations building or proposed and represent anticipated performances. On the three ships actually in service anticipated results were in every case exceeded.

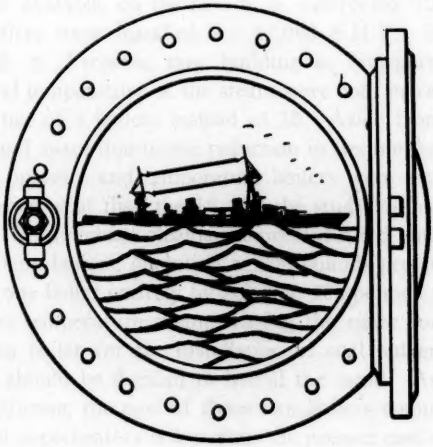
One of the advantages from a marine point of view of higher pressure and temperature installations is the saving in weight and space possible. The turbines are more compact and the saving in boiler heating surface for a given amount of power is very appreciable. For example, on the new S. S. *California,* 12 Babcock & Wilcox boilers were installed for 17,000 S.H.P. In her sister ship, the S. S. *Virginia,* now building at Newport News, the pressure and temperature of the steam were both increased, resulting in the use of 8 boilers instead of 12. Aside from the saving in weight and space due to the reduction in the number of boilers, the higher pressure and temperature boilers were installed at appreciably lower cost than the 12. In the study we have made for a 3300 S.H.P. merchant steamer at present fitted with three Babcock & Wilcox boilers, operating at 200 pounds pressure, we have eliminated one boiler entirely by going to 400 pounds pressure and 700 degrees temperature steam, leaving the space formerly occupied by this boiler for the installation of coal pulverizing equipment if it should be decided to install the same. As in the case of the *California,* the cost of these two boilers complete with air heaters and superheaters is less than the present cost would be for the three older boilers with superheaters.

Time is not available to go into detailed figures, but for those who care to make a study of the same it may be shown that there is no propulsion equipment available, for marine use, which can compete with moderately high pressure and temperature steam from the points of view of cost of installation, weight, cost of operation, and dependability. Every one of these factors is of importance in the marine field and cannot be ignored by those charged with the responsibility of operating ships in these highly competitive times. Competent engineers now realize that *thermal* efficiency by itself plays only a relatively small part in the econom-

ical operation of a ship, and that it does not pay to secure this high thermal efficiency at the expense of the other more important factors.

The situation in the stationary power plant field clearly points to the path which must be followed to secure the generation of power economically, and shows that for this purpose steam at the higher pressures and temperatures is the answer.

# ANNUAL DINNER
OF THE
# AMERICAN SOCIETY
OF
# NAVAL ENGINEERS



THE WILLARD
Washington, D. C.
March 29, 1928

# BANQUET OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

## THE WILLARD, WASHINGTON, D. C.

MARCH 29, 1928.

REAR ADMIRAL JOHN HALLIGAN, U. S. N., PRESIDENT OF THE SOCIETY.

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REAR ADMIRAL WILLIAM H. STANDLEY, U. S. N., TOASTMASTER.

ADMIRAL HALLIGAN: *Mr. Secretary and Gentlemen.* The American Society of Naval Engineers again welcomes its members and guests to an annual banquet. This Society was organized in 1888, so for 40 years it has exercised a benignant and stimulating control over the rapid development that naval engineering has had during that period. When a small group of officers of the old Engineer Corps organized this Society, naval engineering was very modest as compared with its present conditions. The appropriation Steam Machinery 1888 was $675,000, as compared with $19,400,000 carried in the present bill. The *Chicago,* with 5084 horsepower, was in 1888 the ship with the most powerful machinery plant that had been completed. We have now the airplane carriers *Saratoga* and *Lexington,* each of 180,000. Those of us who have engineering close to our hearts should appreciate what we owe to that group of officers, young and enthusiastic, who gathered in a room in the Navy Department 40 years ago and gave us this Society, a Society which was organized, in the words of Asa Mattice, to promote the prestige and efficiency of naval engineering. And tonight we have with us three of the young men who organized this Society, and in order that you may see them and may salute them, I will ask them to rise for a moment as their names are called.

Rear Admiral George W. Baird,
Rear Admiral Robert S. Griffin,
Rear Admiral Harold P. Norton.

Each year the society holds a prize essay contest, and the award for 1927 has been made to Mr. Henry F. Schmidt for his very excellent paper on "Some Propeller Experiments with Particular Reference to Pumps and Blowers." Mr. Schmidt is consulting engineer with the Westinghouse Company, and has made developments in other directions—an oil governor for turbines, and certain turbine arrangements, and if Mr. Schmidt will come up to the front I will be very glad to give him the Society's gold medal and a check for a hundred dollars.

(The gold medal and check were presented to Mr. Schmidt.)

MR. SCHMIDT: I would like to thank the Society for the honor bestowed upon me, and I assure you that I consider it the greatest honor that I could receive, and I have always wanted to do something to connect myself with naval engineering.

ADMIRAL HALLIGAN: In its present stage of development, naval engineering owes much to the sympathetic and intelligent help that comes to it from line officers in high command. Admiral Hughes, as Commander in Chief of the United States Fleet and as Chief of Naval Operations, has guarded the interests of engineering zealously, and it is significant of this happy condition that Admiral Hughes, while Director of Fleet Training, and his successor in that office, Admiral Montgomery Taylor, should have been President of this Society, and it is of further significance that the toastmaster of this evening is Admiral William H. Standley, at present Director of Fleet Training, and to him I now turn over the meeting.

TOASTMASTER: *Mr. President, honored guests, and members of this American Society of Naval Engineers.* A few days ago I was reading a little treatise on organization, and I ran across a statement to the effect that problems were difficult of solution because of the variable focal distance. The writer of this was speaking of organization, but it seemed to me that the statement was applicable to all problems, and particularly to naval problems. The Navy is not only involved with the application of shore developments, with the mysteries of the sea and the air, but when a decision is reached, we still have the more difficult problem of convincing a sometimes all too skeptical public that they need the insurance that we think they need in the way of national protec-

tion. It is the variable focal distance again. Various ways and means have been devised by the Navy to assist and further the solution of these problems. The work of this American Society of Naval Engineers has been of vital importance in the development and improvement of materiel. Navy personnel has also been untiring along these lines. The Congressional Record of March 24th contains some four and one-half pages of closely printed matter on which is recorded the names and works of the various officers of the Navy. All have been instrumental in the improvement of Navy materiel. The Navy's greatest problem, however, is not material. It is in its contact and responsibility to Congress and to the people. It is here that we find the greatest variation in the focal distance. The fathers must have visualized this situation, because the Constitution and later laws are so worded as to provide for a civilian head of the Navy, whose function it is to bring the Navy and the people into closer contact.

Our first speaker tonight is a man who is particularly well equipped for this responsibility of establishing a common viewpoint on naval questions for the Navy and the Congress and the people; a graduate of the Naval Academy of the class of 1888, who has since distinguished himself in his profession and on the bench in civil life. It gives me pleasure to introduce to you the Honorable Curtis D. Wilbur, Secretary of the Navy.

SECRETARY WILBUR: I do not desire to take very much of your time tonight, a group of technical men gathered together partly to exchange information and partly for social purposes. Perhaps, however, a few words may fit into your program, and I cannot help reminding those who are here of some of the changes in the Navy. We are very much interested in the change in scientific development and equipment. I am glad to know that we have with us Mr. French, the Chairman of the Subcommittee of the House, who struggles day after day and week after week and almost year after year with the Naval Appropriation Bill. I am sure that we all appreciate his efforts. I know that he is trying to be economical, but I want to say that we don't know anything about economy. There was a time in the American Navy when we economized so much that we ran our machinery on a vacuum. The boilers were so run that we took a chance of their blowing in rather than blowing out.

When I left the Naval Academy I went to Los Angeles, and there I met a graduate of the Naval Academy who had retired as an Ensign, a Mr. Massey. Believing that I had had some training, he felt that he had a scientific soul to whom he could confide his ideas and his discoveries. I remember that I felt I was dealing with a man who was a crank, perhaps not all there. But what he was trying to tell me was that in his opinion there was some fundamental unit which mixed in various proportions made the different atoms which gave us our different elements, and kept at that, speaking to me every time he had a chance of some of the correlated ideas, and I thought you would be glad to know that a naval officer was a pioneer along the line which we have seen in the development of the radio and in our scientific development as one of the greatest steps in the development of chemistry and of science—the idea, the conception that the atom is not the smallest subdivision of matter, but is composed of other things.

The Navy has grown. At the close of the Civil War the horsepower of all the vessels of the Navy was 475,000. The midshipmen, our naval cadets as they were called at Annapolis at the time I was there with Admiral Hughes, Admiral Robison, Admiral Wiley, General LeJeune, and others, were given the things that were left over from our wars. I do not remember anything from the Revolutionary War unless it was some of the cutlery, but the *Constellation* was a product of the Washington administration, fought in the War of 1812, and was kept in repair for the training of midshipmen to use the instruments of modern warfare. We also had some things left over from the Civil War, the old *Wyoming* and the *Passaic*. But these were rather valueless and a new Navy was started, and in 1898, when the Spanish-American War came upon us, the horsepower of our new Navy was 395,000, a little more than the combined horsepower of the *Lexington*, the *Saratoga* and a destroyer. Last year we ordered airplane engines with a total horsepower more than half the total horsepower of the entire Navy in 1898—240,000 horsepower of airplane engines ordered in one year. The 71-ship program called for a horsepower of three million, over eight times the total horsepower of the Navy in 1898, and the revised program with 16 ships only calls

for a million and a half horsepower. So we are moving forward. I am taking a moment to remind you of Sprague and his motor introduced on the street cars shortly after my graduation. I think the first street car line was in Richmond, Va. Los Angeles, Cal., had one about the same time when I arrived there in 1888, with two overhead wires and a four-wheel trolley, finally taken down as useless.

We have a great deal to say about the progress of engineering and the contributions of the engineers, but I think tonight as I speak to you of an old man 72 years of age, bowed in grief in the city of Los Angeles, Mulholland, who after developing that magnificent water system, erecting 19 dams, found to his horror that one had failed and hundreds of people had lost their lives, and on the witness stand he took the blame and said: "I must have missed something somewhere." So it is true sometimes we are heard of because of our mistakes as well as our advances and our successes. I am glad to say that we have gone forward in the Navy with new things, that our engineers have been doing their part, and that accidents of a major character are largely disappearing because of the scientific care and achievement of the engineer officers of the United States Navy.

I merely wanted to extend to you a greeting tonight, and to express my appreciation to the members of our Appropriation Committee who are here, for their splendid work, and to the other members of Congress who have sustained the Navy for years, in an effort to make a Navy that shall be a credit to the United States of America.

TOASTMASTER: I am sure that you will all agree with me that it is always interesting to hear of the old Navy and of the things we used to do in the past.

There is another branch of the Government service which is also vitally concerned with the variable focal distance. Members of Congress, in their efforts to solve the problems for their constituents and the country, must go out into the byways and hedges searching for the light. Concerning naval problems some of them consult experts of our own country. Some of them not only consult experts of our own country, but they consult the records and information of every other country. Some of them not only con-

sult the records of our own and every other country and all the other information they can gain, but they never miss an opportunity to gain first-hand information of the Navy and its needs by personal association and contact with its personnel and their work. Our next speaker is one of these, and it gives me pleasure to introduce the Honorable Peter Norbeck, Senator from South Dakota.

SENATOR NORBECK: *Mr. Secretary, Mr. President, and Members of the Society.* A good many years ago a neighbor of ours served on a jury. He had had no experience in that line, and he was quite impressed with the fact that he had been called upon for such important work. After the case had been disposed of he went to the attorney and asked how it was that he had been chosen, and he was told that they wanted a man who knew nothing about the subject and whose mind could easily be changed. I felt a good deal like that when I was selected to the Naval Affairs Committee of the Senate, and it was with a good deal of hesitation that I accepted. I was thinking of naval construction, cruisers, battleships and destroyers that I had read about and never seen. But at the first committee meeting I was there, and the first question we were called upon to consider was whether the Government should reimburse a certain officer for the loss of his baggage. If anybody wanted a promotion he would ask Congress to help him in that line, and before we adjourned we had another case. One who had got tired of the Navy and resigned wanted to get back in, and he wanted an act passed for his benefit. I began to see that work on the Naval Affairs Committee on the seaboard was a good deal the same as on the Great Lakes—how to avoid collisions. We hear a good deal about disasters that have befallen some of the ships now. When we get that solved we are going to try to prevent collisions on Pennsylvania Avenue. I suspect that one will be about as easy as the other.

I come from a section where we know less about the subject that you are interested in because we are farther away. There is a conflict of opinion. But we had people who supported Theodore Roosevelt when he pleaded and pleaded in vain for a ship or two. But it is a long ways from such a program to the one we are wrestling with now. Of course, we have people who believe that the Navy is the cause of war; that there is a chip on our shoulder,

and any one who is looking for trouble always gets it. Every member of Congress from the interior of the country gets letters saying that the way to reach a good understanding is to abolish the Army and destroy the Navy. We made a start a few years ago toward destroying the Navy. Some of those who worked for that program are not so sure as they were then. But we have the same ideas and the one desire, the desire for peace and a better understanding, and each has its argument. Some say that with 3000 miles of border line, without a soldier, without a battleship or a gunboat, we have had peace for a hundred years, and why can't other nations do the same? Possibly for the same reason that we have never succeeded in doing the same thing to the south of us on the Rio Grande. Maybe it doesn't lie in the form of government that the republics to the south of us have. And still we have peace and good will, and we cannot claim all the credit ourselves for this. A high official from one of the republics to the south of us visited this city. He was entertained royally, shown around Washington, visited the different departments of the Government, but he had a sort of a feeling that maybe he hadn't seen it all. So one morning early he went out through the byways of the city, and when he got ready to leave he was asked what impressed him the most of anything he had seen, and he said the fact that you can leave a milk bottle outside from four o'clock in the morning until seven without having it stolen. With all the milk bottles outside there will be less misunderstanding, there will be less interference, but until we reach that time we cannot disarm and we must always be prepared.

I came here tonight with two promises, one that I would get a good dinner, and the other that I would not be asked to speak very long. I have had a good dinner, and I thank you for the invitation, and hope that at some future time I may have the pleasure of being with you again.

TOASTMASTER: I am sure that we all appreciate very much the remarks and views of Senator Norbeck concerning the Navy and Congress, etc. And there is one thought that I think we can take home with us, and that is that human nature is the same whether it is in Congress, in the Navy, or out on the plains.

We also have with us tonight another distinguished Naval Academy graduate from civil life, a man who was a success as a student and an athlete while at the Academy, and as an officer of the Navy, and who has been conspicuously successful since leaving the service. A man who never was so successful, however, that he could not find time to retain contact with the Navy and interest in its personnel; a man who has done much to help the people to a better understanding of their Navy. It gives me great pleasure to introduce to you Mr. Joseph Wright Powell of Boston.

MR. POWELL: *Mr. Toastmaster, Mr. Secretary, Guests and Fellow-members of the Society.* When I got John Halligan's telegram one Sunday asking me to speak here, the first thing I said to myself was he certainly is hard up for speakers. But I accepted, for I had something on my chest that was worrying me. Now I would not dare to talk to you gentlemen about naval engineering. Some seven years ago I left the profession of active shipbuilding. Mat Brush was up in Boston a few days ago to see some of his old friends. He said that in 1916 he was the smartest manager of an electric railway in the United States, and he said he could prove it because he got out of the job in 1916. Now I claim the same distinction as the smartest shipbuilder in 1921 because I got out in 1921. But engineering and shipbuilding have changed to such an extent that far be it from me to tell these people anything about their business. So I wrote to John Halligan, and said: "How long do you want me to speak?" and he wrote back and said: "You can speak 20 minutes," and he underscored the 20. Now that reminds me of a story about the new Ford. A man driving a big Packard along the highway heard the chug, chug, and he looked out and saw a Ford, so he stepped on the accelerator and speeded up to 40. In a minute he looked out and saw the Ford again, and then he went up to 60, and pretty soon the Ford was alongside of him, and then he went up to 80, and soon he heard the chug, chug, toot, toot, of the Ford, and he looked out and the Ford driver said: "Say, mister, how do you get this car out of second?" If I am to get through in 20 minutes, I have got to get out of second.

I want to talk to you a little about the Naval Academy tonight. Our first American midshipmen served in the British Navy and

not until our wars with France and the Mediterranean powers were midshipmen appointed to the United States Navy. The Navy Department was organized in 1798. Two years later a school for midshipmen was nearly included in the arrangement for the establishment of West Point, the Military Academy actually beginning its career in 1802. But midshipmen were not destined to receive any schooling at the hands of the Government, except in the school of experience, until 1813, when the first schoolmasters were authorized. Three years later the Secretary of the Navy was directed by Congress to report a plan for a Naval Academy, but nothing came of his recommendation, and six years later a similar fate befell a proposal to Congress by the Secretary of the Navy, and it was not until 1833 that paid instructors were provided by Congress at Boston, New York and Norfolk, who gave instructions to such midshipmen as wanted to take it, but not until 1838 was a real attempt at a Naval Academy begun. Up to that time naval education did not amount to much, as illustrated by the old Navy yarn about Midshipman Tatnell's examination. A Commodore, who was the head of a board, looked over the young gentleman and said: "Mr. Tatnell, you command a ship on a lee shore, wind blowing a gale, both anchors lost, all sails blown away, all boats stove in, ship scudding rapidly toward breakers, what do?" Midshipman Tatnell promptly replied: "Unlace my shoes and trust to God."

In 1838 the first real attempt at a school for Midshipmen was made at the Naval Asylum in Philadelphia, perhaps some significance in the conjunction of forces, and four years later William Chauvenet was in charge of the school. I think he really deserves the title of father of the Naval Academy, because he worked out a scheme to get a Naval Academy without asking Congress for an appropriation of money. In 1845 Mr. Bancroft became Secretary of the Navy, and he accepted Chauvenet's plan, and eighteen months later our United States Naval Academy was established at Annapolis, Md., in old Fort Severn. Bancroft was a shrewd politician, for he obtained a favorable recommendation for the Chauvenet proposal from a Board of Line Officers before he put it in operation, thus insuring its acceptance by the Navy. The new school ran for more than a year on appropriations made for the

pay of schoolmasters until in 1846 Congress was prevailed on to appropriate $28,000, being the sum that had been annually expended for schoolmasters up to that time, and this was the real beginning of the Naval Academy.

I have cribbed most of the above facts from Park Benjamin's History of the Naval Academy, which is a delightful book which most of you in the Navy have read. It traces the growth and the varying fortunes of that school down through my own days at the end of the last century. It is very interesting to see the struggles that it went through, the difficulty of obtaining any recognition from Congress or very much help from the Navy Department, and the varying fortunes and the changes in discipline as the ideas of one superintendent or another prevailed. Park Benjamin, in his introduction to his History of the Academy, says: "The task of writing this volume might better be done by one wearing the uniform of the United States than by one who doffed it more years ago than he cares to remember, it is needless to say. Nevertheless, an abiding affection for all that pertains to the Navy and a deep sense of his obligation to his Alma Mater may, perhaps, be pleaded by the author for some qualification for his present attempt." In what I have to say tonight I plead these same extenuating circumstances, and my criticism is only called forth by a deep desire to serve the institution, to which I give full credit for such happiness and success as I have achieved in a busy and interesting life.

At the present time the Naval Academy is fortunate in having as its superintendent the man who is best fitted by experience and by ability to head that institution. I think the administration that he is giving is everywhere considered as excellent to a degree, but for causes entirely beyond his control, it is impossible to-day to turn out officers who are as well fitted to meet the problems of the modern Navy as were those who came from the Academy in its earlier days, when the tools with which they had to work were simplicity itself in comparison with a 1928 Navy.

Education does not stand still. Available fields of knowledge grow apace. Eighty-three years ago when the Academy was founded it was comparatively easy in a four years' course to acquire the knowledge that was necessary to equip a young man to satisfactorily meet the requirements of the then Naval service.

Think how things have changed since then! When we studied electricity at the Academy 32 years ago, Thompson's Dynamo Machinery comprised most of the then accepted knowledge. If to-day's electrical science is represented by the alphabet's 26 letters, what can be contained in two similar volumes would represent less than the letter A.

In 1923, when I had the honor and pleasure of acting as a member of the Board of Visitors at the Academy, I asked Dr. Angel, the President of Yale University, whether it was possible, with modern methods of teaching, to put any more knowledge into a boy's head between 16 and 20 than they could when I was a midshipman, or when the Academy had its beginning, and he replied: "Mr. Powell, the capacity of the human brain has not increased in a thousand years." A naval officer must be a steam engineer, an electrical engineer, a navigator, a radio expert, an ordnance officer, and more recently, an aviator. He must know international law, should know foreign languages, and the range of knowledge that should be his is almost beyond the capacity of any one man to achieve. Nowadays you do not make a lawyer in four years, it takes seven, and then he learns only one small branch of the law. A doctor must study for at least seven years; to get a degree in business administration requires six years of intensive work, and yet we are trying to make a naval officer in four years. At a technical institution such as the Massachusetts Institute of Technology, the four years course merely produces a man who has a degree in some particular science. Those degrees to-day are really elementary, for a man who is to master any particular subject must go back for advanced courses, and perhaps for several advanced courses in several different institutions.

In 1923 the Board of Visitors at the Naval Academy reported that mathematics were slighted, that French and Spanish suffered from inadequate time devoted to them, that the English courses were lacking in essential details, and that little more than rudimentary ordnance, electricity, chemistry, physics and radio and aeronautics could be included in the crowded curriculum. When I found that the graduates of the Naval Academy who are going to take post graduate courses in other institutions had to go back to the Academy for a year of special mathematics before they were

sufficiently advanced to take post graduate courses, I was shocked to a degree. The Board of Visitors in 1923 recommended that, in view of the fundamental changes in educational problems presented to the Naval Academy by these extraordinary developments in modern science and technology, and, further, of the necessity of essential reorganization in the work of the Academy to meet these changed conditions, that the Secretary of the Navy consider the appointment of a commission containing representatives of the important branches of the Navy, of civilian education, and such other interests as he may deem appropriate to make an intensive study of the work of the Academy, and to report recommendations for enabling the Academy to more adequately meet the demands of the Naval Service. I believe such a board was never appointed, and so far as I know no such work has ever been adequately done. I believe that the results of such a study would lead to recommendations for radical changes in the present course to make the four years spent at the Academy far more foundational; to give more time to foreign languages and mathematics, and much less time to technical studies. You cannot build a sound education on an incomplete foundation, and that is being more and more recognized by all of our great technical schools. Now education of the Naval Officer must go on through life. It seems to me that such a board would probably recommend that after appointment as provisional ensigns and a short cruise at sea, one or two years, that every graduate would return for two years of highly scientific technical work, and in my own judgment no less than six years of such study can entitle a man to a commission, a full commission in the United States Navy. There are many other recommendations that such a board might make. It seems to me, for instance, that some outside control such as exists in all universities might be very beneficial; that a Board of Trustees, composed of course largely of the Navy, but with some outside representatives, that had a hand in the selection of Superintendents and in other matters in connection with the curriculum, would do a great deal to stabilize the work at the Academy.

Now if such a Board were organized through the demands of the service, I think that its findings might be of real value. A member of the Board of Visitors, who had served on several

boards, said to me the other day that his work was unique in that no single recommendation that had ever been made by any board on which he had served had ever received the slightest attention. Now I think that one thing that might be done would be to abolish this Board of Visitors. It takes up a good deal of time at the Academy, and its services are worthless. It is moreover quite fruitless to undertake such an investigation as I have proposed unless and until there is a background of demand for better educated graduates who can more fully meet the technical requirements of an exacting service. One cannot look through a modern ship to-day, the *Lexington* or the modern submarine, without realizing the complexity of these instruments of war. To put these vessels into the hands of officers with incomplete engineering training who have had no proper chance for the technical study necessary for an appreciation of the forces with which they are dealing, seems to me a gross injustice to these young men, and I believe it calls for urgent and immediate attention by the older officers of the Navy. Nor is it outside the facts that such better education will go a long ways toward eliminating some of the accidents that have caused to those who have left the service the deepest concern. It is with this firm belief that I have come here tonight. It has been a real pleasure to get back and see you all again, many of whom I have not seen since I spoke before this Society seven years ago. I hope it won't be that long before we meet again. If anything I have said is seed sown on fertile ground I shall feel a thousand times repaid.

TOASTMASTER: I am sure you will all agree with me that Mr. Powell has fully come up to my statement to the effect that he has never ceased to work for and with the Academy and the people of the service. The information that he has given may well be taken home by many of us and given careful thought and consideration. We all have the interest of that Alma Mater at heart, and anything that will make that institution better and bigger and broader is well worthy of our thoughts.

We are living in an age of great scientific development. The gas engine and the radio have affected every form of industry and have practically revolutionized transportation and communication. A message catapulted into space to-day travels several times

around the world before it finally fades away. Men are flying through the air almost as freely as the birds, and the end is not yet. We can put almost anything into the air, but we can only take out as yet that which we put in. I predict that sometime in the future instruments will be perfected which will react to air waves, the nature and extent of which at the present time we have not the slightest conception of. Our next number on the program will indicate to you somewhat the trend along these lines. The gentleman has chosen for the subject of his interesting demonstration, " Narrow Casting, or making light audible and sound visible." I take pleasure in introducing Mr. Taylor of the General Electric Company.

MR. TAYLOR: *Mr. Toastmaster, Naval Engineers, and honored guests.* The committee have been good enough to list me with the speakers and give me a seat at the high table, but it is the apparatus set up on the floor rather than what I may say that will interest you. Nevertheless, I am allowed a few minutes to tell you about it so that you may better understand what you are to see and hear:

At the present time, in addition to the time-honored telegraph and telephone, we have the radio telegraph, radio telephone, radio broadcasting, photographs by wire, photographs by radio, motion pictures by wire, motion pictures synchronized with sound, and even motion pictures by radio accompanied by the appropriate sounds. There is such a variety of devices in regular use and under development that a layman may become confused and give up trying to understand underlying principles, and be content merely to turn the dial, listen and look at what is presented for his edification or education.

Now the purpose of all communication devices, of course, is to bring something from the initial mind—the gray matter within the bony box—to other minds located either in proximity or at a distance. We place great dependence on the spoken word, and the spoken word is such a common thing that we accept it without pausing to consider the remarkable chain involved in carrying an idea from speaker to conscious mind of listener. In the first place there must be (or should be) an idea in the brain before an attempt is made to speak. Nervous impulses, probably of an electrical nature, emanating from the brain go to the muscles of the lungs which pump air through the larynx; co-ordinated impulses

go to the muscles of tongue and lips which modulate the sound made by the air as it is forced through the larynx. This variable air pressure, started by the speaker, spreads and may reach several hundred ears as receiving stations.

In "radio broadcasting" we have all the above links and in addition a few more. The energy, instead of passing directly as an air wave from speaker to listener, goes through translating and transforming devices in which the mechanical energy goes into the form of electrical energy, and is as a wave, launched into space. Space has the remarkable property of carrying these electrical impulses at a velocity a million times greater than the sound wave moves through the air. This tremendous increase in velocity means little when dealing with short distances such as a hundred feet or more in an auditorium; but since our electrical wave can be received, detected and amplified after 10 miles, a thousand miles, or even at the antipodes, the element of time becomes of real importance. Suppose means for producing sounds of sufficient intensity, with suitable speaking tubes and receiving devices for communicating between New York and Chicago: a person could say "hello" and go out and get dinner before the answering "hello" was heard. In other words, sound is altogether too slow for conversations over considerable distances.

Now, remarkable as the broadcasting art has grown to be, there has always been, since the beginning of radio communication, a desire for restricted communication—at times because communications are confidential—at other times where the confidential element cuts little figure, in order to avoid the tremendous confusion that results from being obliged to hear much that you are not interested in along with the desired message.

Directing the radio wave has met with some success, increasing as the wave length has been progressively reduced. For short waves the antenna can be constructed so as to give a directed and somewhat restricted beam which not only makes transmission more efficient but also reduces interference. However, as yet, such beams cannot be directed and restricted sufficiently to give the desired element of secrecy. Radio wave lengths are expressed in meters and shortest radio waves in use are numbered as tens and units. But light waves are expressed in fractions of a millionth of a meter. The "radio" wave and the light wave are of the same nature—the difference is one of magnitude.

It is therefore a logical step, where an accurately directed and closely restricted communication beam is required, to jump to the light wave. A different form of generator serves for this radiation. An incandescent lamp is a generator of radiation just as much as a broadcasting plant is, and as soon as we pass to light radiation as distinct from the longer wave radio work, it is a simple matter to direct a narrow beam by lens or mirror of moderate dimensions. There is no chance to tap or intercept unless a device is placed directly in this restricted beam. Since interference is reduced to a minimum, an unlimited number of communicating channels are available.

There is usually, perhaps always, some loss to offset a gain. A beam of light will not pass through a fog nor smoky atmosphere; it will not bend around obstacles, and, compared to "radio," the working distance is restricted.

Bell experimented with and described his "photophone" about 50 years ago. Since Bell's early work we have become familiar with amplifiers, loud speakers, improved incandescent lamps, and other devices which I have incorporated in the "narrow-casting" apparatus set-up for demonstration with the sending and receiving stations at opposite sides of the room. The term "narrow-casting" has been used because it has common features with radio "broadcasting" and suggests the difference from the now familiar art.

The apparatus was then described and demonstrated. At the sending end an oscillograph mirror reflected and deflected light received from a small concentrated filament incandescent lamp into a lens which focused the light in a narrow beam to the distant receiver. The oscillograph was connected to an electrical "pick-up" on phonograph record; in action the oscillograph mirror followed the vibrations of the needle and pick-up of the phonograph and "modulated" the beam of light in correspondence with the music.

The receiver consisted of a photo-electric cell and several stages of amplifying tubes leading to a loud speaker. When the beam of light as modulated by the record was directed into the photo-electric cell, the music was heard by all present. The beam was carried over the length of the speaker's table; the music could be stopped by placing the hand or any other opaque object in the beam at any point between transmitter and receiver; the volume of

the sound could be controlled from full intensity to silence by regulating the extent of beam intercepted.

Demonstration was also given of the response of the receiver to different kinds of light. A 60-cycle incandescent lamp gave a low-pitched even tone; a portable lamp with hand-operated generator sounded like a siren; a match when struck gave explosive crackling sounds.

For the transmission of speech a diaphragm with tilting mirror may be substituted for the oscillograph; or a microphone conveniently located may be electrically connected to the oscillograph.

# DISCUSSION OF MR. SCHMIDT'S PAPER ENTITLED "SOME SCREW PROPELLER EXPERIMENTS WITH PARTICULAR REFERENCE TO PUMPS AND BLOWERS."

PUBLISHED IN THE FEBRUARY, 1928, ISSUE OF THE JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

BY H. F. HAGEN, MANAGER OF RESEARCH, B. F. STURTEVANT CO.

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The paper hardly justifies Mr. Schmidt's implied criticism of standard propeller fan analysis. By standard propeller fan analysis I mean the analysis of the flow of air and the action thereon by a propeller fan that has grown up in the literature of the subject, starting from the original contribution of Euler and continued by other writers unchanged to the present day.

In the first place I have considerable difficulty reconciling Mr. Schmidt's theory to which he refers in his opening paragraphs. I do not see the connection between his theory and what he has done in his design. He says, properly enough, that the vena-contracta of a sharp-edged orifice is 62½ per cent, to which we all agree. As a matter of fact neither Mr. Schmidt nor anyone else would use a sharp-edged orifice as an entrance to a propeller fan. Some sort of cone or rounded entrance would certainly be used, thereby eliminating the action of the sharp edged orifice. If Mr. Schmidt had used a sharp-edged orifice for his propeller fan and had made the outlet diameter of his blades smaller than the inlet diameter in such ratio that his outlet wheel area was 62½ per cent of the inlet area of the wheel, there would seem to be a logical sequence in his theory. As a matter of fact at one time Rateau did make his wheel to operate right in a continuation of the inlet cone, thereby approximating what I thought Mr. Schmidt was going to do.

As near as I can understand it, Mr. Schmidt considers that his number, 62½, has some cabalistic property and uses it to de-

termine the projected area of his blades. In other words, he uses the same inlet and outlet diameter on his blades, but only utilizes 62½ per cent of the blade circle. It does not seem that this has anything to do with his theory, nor do his tests show that it is particularly important in the results. His Figure 9 on page 8 shows no measurable change from 0.4 projected area ratio to full projected area ratio. If we examine Figures 19 and 20 we find that the first of these shows a propeller with 63½ per cent projected area ratio and the second with 100 per cent projected area ratio. The comparison is substantially as we should expect, a greater capacity from the wider blade fan. Slight differences are shown in efficiencies, but the accuracy of the test is such that these differences cannot be considered real.

Sharp-edged orifices were used to measure volumes. The flow through a sharp-edged orifice is probably the most unreliable of any of the measuring methods that might have been selected. The author states that the velocity of approach to the orifice was, to all practical purposes, zero. The velocity in a room in which a fan is acting will never be zero; and even slight variations in the direction of the velocity of approach to the orifice will greatly change the co-efficient. When the whole apparatus was so small it would have been better to have used an intake pipe and a pitot tube measurement of the volume made in the pipe; or properly formed nozzles might have been used. The expense of these small nozzles would have been inconsiderable and any doubt about the volumes would then have been eliminated.

There is not found in the article a complete description of just how the pressure readings in the box were taken. Apparently these were merely readings of the static pressure in the box, which is an incorrect method for such a set-up. The incorrectness of the method is evident from the author's own words on page 13, the first sentence beginning on that page, in which he states that he obtained efficiencies, before he put in what he considered proper baffling, of the order of 120 per cent. It is axiomatic that the 120 per cent efficiency is impossible. The author's explanation that the velocity was helping the fan is bad science. Under no conditions could the efficiency be over 100 per cent. Some other explanation must be found. It is, I believe, indisputable that the test

methods were incorrect. There seems little doubt that the error lies in the interpretation of the pressure measurement in the box.

A brief example, with assumption of ideal conditions, is probably the best way to expound the error. Let us assume a flow without losses. Let the velocity of the air stream into the box be 4,000 feet per minute, which corresponds to 1-inch velocity pressure. The static pressure in the box will then be — 1 inch. If this is credited to the fan, the resultant false efficiency might well be over 100 per cent. If, however, an impact tube were placed heading into the air stream, it would show under our ideal flow conditions the sum of the minus 1-inch static pressure and the plus 1-inch velocity pressure, or zero. Under actual flow conditions, with friction and eddy losses, this impact tube would show the loss which might be only a ¼ inch or so. Only this amount can be credited to the fan.

In any test of the type described the impact tube should always be used; and, if used, would give a correct indication of the fan efficiencies under any particular conditions of baffling which might exist. Efficiencies would then indicate true differences introduced by changing the inlet flow by different baffling; but absurdities such as over 100 per cent would be impossible. The correct method of making a test as described is completely given in the code of the National Association of Fan Manufacturers, and by following this code exactly the trouble experienced by so many investigators in distinguishing between total and static efficiencies will be avoided.

Mr. Schmidt's arrangement of the baffling after discovering this 120 per cent efficiency was not scientific. Apparently he first put in a lot of baffling, which reduced his efficiency below what he finally reported. He goes on to say, "If the air is completely brought to rest by a too extensive baffling system, the efficiency observed will be less than the real efficiency, and if the baffling is insufficient, too high an efficiency will be obtained." In the first place, it should be apparent to anyone that the cross-sectional areas in the box are finite and, therefore, with a finite volume passing through the box no arrangement of baffling could possibly reduce the velocity to zero or bring the air completely to rest.

An intermediate degree of baffling was then used. The author tries to justify this intermediate degree of baffling by talking about

a natural flow to the fan wheel. I have always taken the stand that propeller fans are so sensitive to the flow that comes to the inlet that they are very liable to be undependable in actual installations. On this point the author of the paper and myself are in agreement. I cannot but feel that Mr. Schmidt's undoubtedly earnest endeavor to approach what he calls natural conditions of flow, when he was determining the intermediate baffling, was perhaps subconsciously colored by his desire to secure a high efficiency. Apparently, by the arrangement of his baffling he had the choice of between 45 per cent and 120 per cent efficiency. I respectfully submit that the tests, as reported, more nearly represented the peculiar result of the intermediate baffling on the pressure in the box than they did the efficiency of the fans.

Before leaving this subject let us refer to the curves in Figures 25 and 26, on pages of the same numbers respectively. These tests were run to determine the effect of the number of blades. So far as I can see from the data given on the curves, the pitch, projected area ratio, guide vanes, etc., were the same in all cases. Consider the wide open volumes shown on these curves. In Figure 25 three blades shows approximately 3100 C.F.M. and four blades 3050 C.F.M. In Figure 26 five blades shows better than 3100 C.F.M. and six blades shows about 3010 C.F.M. In other words, three blades and five blades show better than four blades, and six blades worst of all. If this were represented in a curve it would show a minimum, or a depression in the curve, at four blades and a substantial equality between three blades and five blades. I will admit that the differences in every case are small; but it is just such differences that Mr. Schmidt relies upon to establish the desirability of his 63½ per cent projected area; and it is just such differences that are shown in his Figure 9 to which I referred before.

The above criticisms are rather insignificant compared with the criticism of the usual theories which Mr. Schmidt makes on page 15, the first paragraph starting on that page, in which he uses the word "fallacies." Without any justification other than his own observation of some smoke given off by a piece of smouldering waste, he concludes his paragraph with the startling statement, "These observations show the fallacy of the idea that in a pro-

peller type pump or blower the air or water is whirled by the propeller and is thrown off the tips."

With regard to the last phrase about the air being thrown off the tips I have no criticism. This condition will only exist when the fan is operating at very low volumes, that is, after you get the break in the pressure curve and start up on the centrifugal type curve. When the fan is operating wide open, or while the pressures are still low enough for it to continue to act as a propeller, I have no complaint to make against the statement that the air flows through in proper stream lines. In fact I believe that such an action actually does occur.

However, the statement that there is no whirl is contrary to fundamental economics. We know that action and reaction are equal and opposite in direction. We know that the only force supplied by a motor through the fan impeller is rotative and can be completely stated as torque. This motor torque supplied to the fan impeller must be compensated by a reaction of the air on the fan impeller equal and opposite in direction. There is no force which the air can exert on the fan wheel other than one due to the reaction from mass $\times$ acceleration. This acceleration necessarily must be a rotative acceleration. The summation of mass $\times$ rotative acceleration $\times$ radius must equal the torque supplied to the fan wheel by the motor. It follows that the air passing through a propeller fan wheel, or any fan wheel for that matter, must receive rotative acceleration and leave the fan wheel with a whirl. This whirl may be taken out by guide vanes; but I submit that any fallacy in connection with this is on the part of the one who claims to have designed a fan in which the air does not whirl.

We, ourselves, favor inlet vanes on propeller fans by means of which we spin the air against the rotation of the wheel. The necessary rotative acceleration that the wheel must give the air is then used to overcome the negative rotation which has been imparted by our inlet vanes with the result that, if our blade angles as actually designed are correct, we can deliver air from the wheel without rotation.

Mr. Schmidt seems to think that if inlet vanes are used they ought to be movable. It is perfectly true that inlet vanes can be designed for only one condition of flow, i.e., for one orifice. The

same thing is equally true, however, of the fan impeller. Its angles are correct only for one orifice. Exactly the same thing is true of diffusion vanes, or vanes placed after the wheel. They will have their best result only at one orifice. Such things are not primarily a matter of opinion, although different designers may arrive at different decisions. Such things are decided more by accurate test. If it is found that with stationary inlet vanes the efficiency keeps up to a high point over a broad range, there is no particular need to vary the angles of the inlet vanes.

The same thing is true of guide vanes placed after the wheel. Mr. Schmidt drops a detailed discussion of designing these guide vanes by making them more or less of a paddle-wheel construction. This, of course, is a very evident sacrifice of the highest possible efficiency to secure broader action. The use of guide vanes as Mr. Schmidt uses them on any propeller fan cannot avoid resulting in a wonderful siren. The noise made by vanes so placed is unbearable and uncorrectable.

Guide vanes placed in the inlet, however, particularly if made in a form to resemble a multiblade centrifugal wheel, do not apparently increase the noise made by the fan. Rather they increase its efficiency and the pressure that the fan can develop at a given tip speed, thereby actually decreasing the noise for a given duty.

# NOTES.

## H. M. RIVER GUNBOATS FOR CHINA.

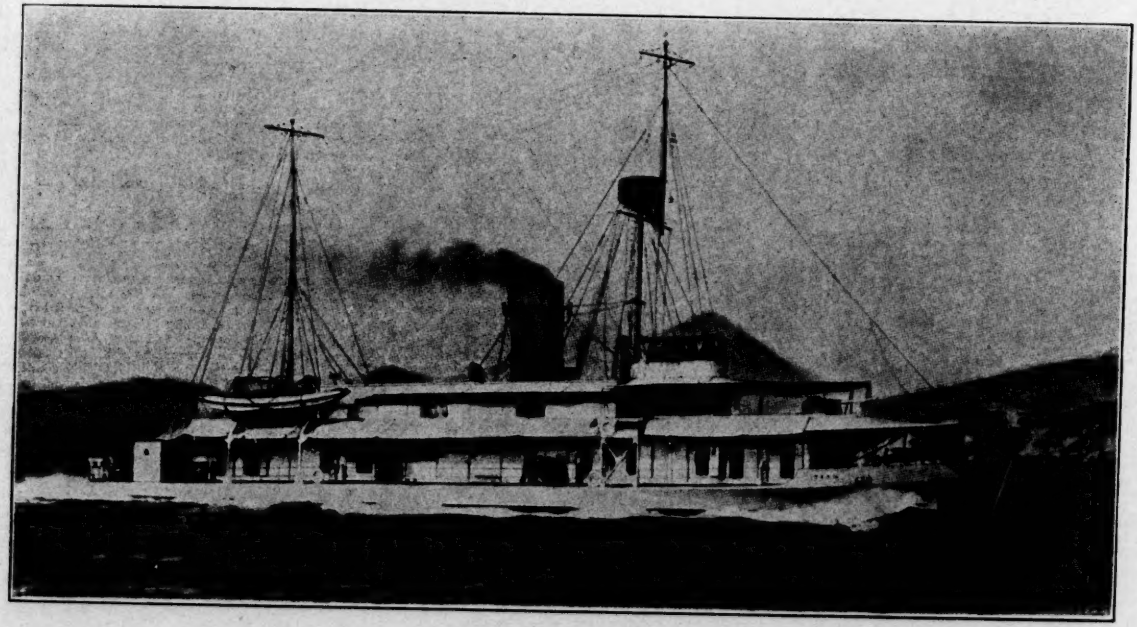
As is well known, the British Admiralty have had in service for many years on the upper Yangtse-Kiang shallow-draught gunboats which have proved very successful in use on this difficult and dangerous part of the river. The section between Ichang and Chungking, on which these boats operate, is extremely difficult to navigate, there being numerous rapids, shoals and rocks, with a very swift current. The conditions require very shallow-draught boats, and it was for this service that Messrs. Yarrow and Company, Limited, of Scotstoun, devised the tunnel system of propulsion combined with the Yarrow balanced flap, an arrangement which was first embodied in H. M. S. *Widgeon*, some 25 years ago.

The same firm has recently supplied, to the order of the Admiralty, for service in the same waters, the gunboats *Gannet*, *Peterel*, *Seamew* and *Tern*, the last-named being illustrated. These vessels are all of the same type, but the *Seamew* and *Tern* are slightly smaller and not quite so fast as the other two. The two smaller vessels have a length over all of 167 feet 9 inches, and a length between perpendiculars of 160 feet. The beam moulded is 27 feet, and the depth moulded, amidships, 7 feet. The machinery, in this case, consists of single-reduction geared turbines developing 1370 shaft horsepower at 450 R. P. M. Steam is supplied by two Yarrow water-tube boilers, each having 1530 square feet of heating surface and fitted for burning oil. These two boats were designed to give a contract speed of 14 knots.

In the case of the larger vessels, the *Gannet* and *Peterel*, the length over-all is 184 feet 9 inches, and between perpendiculars 177 feet. The beam moulded is 29 feet and depth moulded, amidships, as before, 7 feet. These two boats have single-reduction steam turbine installations, developing 2250 shaft horsepower at 400 R. P. M., and the two oil-burning Yarrow boilers fitted have each a heating surface of 2500 square feet. The contract speed, in this case, was 16 knots.

In all four vessels, the armament consists of two 3-inch breech-loading guns and eight 0.303-inch Lewis guns, the three breech-loaders and six of the Lewis guns being mounted on the battery deck, and the remaining two Lewis guns on the fighting top. The accommodation provides quarters for the commander, three officers, 14 petty officers, a European crew of 40 and a native crew of nine. Accommodation for the European crew is on the battery deck, the rest being all on the main deck. Searchlights, electric lighting throughout the vessels, ventilation and wireless service are supplied with current from a steam-driven Peter Brotherhood set, and an oil-driven set run by a Gleniffer motor.

The delivery of these vessels was effected in a somewhat unusual manner. Commonly, light-draught vessels are built up in this country in floatable sections, and, after launching and the completion of trial trips here, the vessels are dismantled and the sections shipped out for assembly at their destination. This method is costly, the sections are liable to suffer distortion, and the system imposes limitations on design. In the present instance, with the co-operation of Messrs. Butterfield and Swire, of Hong Kong, Messrs.



H. M. River Gunboat "Tern."
Constructed by Messrs. Yarrow and Company, Limited, Engineers, Scotstoun.

Yarrow and Company were able to undertake delivery of the vessels complete at Kong Kong, the trials being run there instead of in this country, and the vessels being subsequently handed over to the Admiralty authorities. The boats were, therefore, merely erected at Messrs. Yarrow's Scotstoun yard, dismantled, packed, and shipped, and re-erected and completed at Messrs. Butterfield and Swire's Taikoo Dockyard, the builders remaining responsible until the final handing over after the trials. Re-erection and completion were carried out by the Taikoo Dockyard and staff, under the supervision of Mr. White, representing Messrs. Yarrow and Company, and of officers appointed by the Admiralty. The re-erection was completed without any difficulty, and on trial all vessels exceeded their contract speed. These are the first turbine-driven vessels on the Upper Yangtse.—"Engineering," March 16, 1928.

## SMALLER BATTLESHIPS IN PROSPECT.

American newspapers received during the past week contain a great deal of matter relative to the British proposals for a reduction in the size and armament of future capital ships. This plan, which was first submitted at the abortive Naval Conference at Geneva last summer and has now been reintroduced to the Preparatory Commission on Disarmament, presupposes a genuine desire on the part of all the leading naval Powers to retrench their expenditure on naval armaments so far as may be compatible with security. We believe this desire does exist among the statesmen and the peoples of each nation concerned, in spite of the obstructive attitude of certain groups which have special reasons for resisting any further diminution in the power and prestige of the national fighting forces. When put to the test the influence of such groups is almost always found to be disproportionate to the noise they make, a recent instance of this being the severe defeat which the "big-navy" party in the United States suffered when Congress ruthlessly cut down the new shipbuilding programme. When that measure first saw the light the world was assured, through the agencies which broadcast naval propaganda in America, that the demand for seventy-one new fighting ships had "nation-wide" support and was absolutely certain to be approved by the legislative bodies. It seemed for a time as if this claim were justified, but gradually, as the true significance of the programme dawned upon the American people, it had to encounter a vigorous and ever-swelling tide of opposition, before which the sponsors of the measure found it expedient to beat a hasty retreat.

To those who have followed the course of events since December, when the programme made its appearance, it is clear that Congress, in striking out all but fifteen of the ships originally demanded, acted in obedience to the nation's will. Carried out in its original form the project would have secured for the American Navy, not merely equality with the sea forces of the British Empire, but absolute paramountcy for the American Flag in every ocean to which it could penetrate. The incongruity of launching such a scheme at the very period when the best brains of the civilized world were seeking for practicable means of curtailing preparations for war, and when, moreover, American spokesmen were foremost in urging us in Europe to turn aside from "the Old World policy of competitive armaments," was so apparent that few intelligent observers on this side of the Atlantic believed that the programme would pass through Congress without undergoing drastic amendment. This confidence has been justified by the event. No reflection is cast on the Washington naval authorities by the assumption that they framed their demands on the principle of asking a great deal more than they

really expected to get. Nevertheless, in spite of having obtained fifteen new cruisers, which, added to those already completed and building, will eventually give their navy a definite lead over all others in ocean-going cruisers, they affect to be chagrined by the fate of their programme, and are therefore scarcely in the mood to give sympathetic consideration to the latest British proposals. These, it will be recollected, provide for a reduction in the standard tonnage of future capital ships from 35,000 to something below 30,000 tons and the restriction of gun caliber to 13.5 inches, instead of the 16-inch limit now in force. At first sight the proposals seem fair and logical from every point of view. If naval armaments are to be subject to further restriction, the first and most obvious step is to begin with the capital ship which constitutes the backbone of naval power. By an anomaly to which we have more than once adverted, the Washington Conference encouraged the building of exceptionally large and costly battleships by fixing a standard tonnage of 35,000 and a caliber limit of 16 inches, yet during the war, which had ended only three years before the Conference assembled, the average displacement of battleships never rose above 25,000 tons. Up to the present only two capital ships have been built in conformity with the Washington Treaty rules, and both are British. The *Nelson* displaces 33,500 tons, the *Rodney* 35,000 tons, the armament being in each case nine 16-inch guns. Although these vessels are understood to be quite successful from the technical point of view, we gather that naval opinion as a whole does not regard them as good value for the money, each having cost about £8,000,000. Be that as it may, the time is approaching when we shall have to lay down more of these huge and expensive vessels unless the Treaty is amended in the interval. Under the replacement rules of that compact we are entitled to build fifteen new capital ships from 1931 to 1939, inclusive; America is authorized to construct a similar number, and Japan nine. As each new ship is completed it will automatically displace an older vessel, until, by the time our fifteen new mastodons are in service, we shall have scrapped eighteen existing ships, beginning with the *Iron Duke* and ending with the *Hood*. This process of deletion is necessary in order to maintain the numerical ratio between the various fleets, but it nevertheless suggests two pertinent questions: first, why destroy ships which may be perfectly serviceable and good for several more years of duty afloat; secondly, why replace each of these ships by a successor of considerably greater tonnage, fighting power, and cost? Both questions admit of simple answers if one is thinking of naval power in the conventional way. But the Washington Conference was summoned, and the Washington Treaty was drafted, with no other ostensible purpose than to put an end to the spirit of competition which had previously inspired the naval programmes of the Powers, great and small. To what extent that purpose has really been achieved still remains to be seen. That the latest British proposals are well calculated to promote it is a statement which cannot, we think, be disputed by any fair-minded person. If, four years hence, the three leading maritime Powers should all proceed to lay down 35,000-ton battleships, each designed to incorporate the maximum sum of aggressive strength as dictated by the tactical views of the various admiralties, then the rest of the world might be pardoned if it smiled rather cynically at this curious sequel to the movement for naval disarmament initiated by the three Powers in question.

It must be said at once that the reception accorded to the British proposals by American naval representatives is frankly unfavorable. They profess to have discovered an ulterior motive, namely, Great Britain's desire to conserve her alleged naval supremacy by retaining, in the *Nelson* and *Rodney*, the two most formidable battleships afloat, while denying America and Japan the possession of ships of equivalent tonnage and armament. We feel sure,

however, that arguments of this kind will carry little weight with those who direct American policy. If the objection were pressed, the British Government would, no doubt, be quite ready to agree to the construction of two *Nelson* type ships for the United States Navy, since our aim is not to haggle over a more or less illusory difference in tons and guns, but to foster that spirit of limitation and non-competition in armaments to which our American friends attach so much importance. So far, we understand, the proposals have elicited no formal response from Washington, where, it is to be hoped, they are receiving the consideration they deserve. While there is no urgent hurry, it is desirable that a decision should be come to well in advance of the year 1931, when the first "replace" battleships are due to be laid down, for it is well known that the preparation of plans for a ship of capital rank involves a great deal of preliminary work which may extend over several years. The British plan provides not only for a reduction in tonnage and armament, but for an extension of the useful "life" of a ship from twenty to twenty-six years. Taken as a whole, its acceptance would unquestionably result in a substantial saving of money to all the countries concerned, without any corresponding sacrifice of relative strength at sea. It is encouraging to find the proposals hailed in Japanese official circles as "honest, practical, and capable of forming the groundwork of an agreement which will substantially relieve the taxpayers." Unless American experts are indissolubly wedded to the biggest ship and the heaviest gun, there seems little doubt that the next Naval Conference will result in a sharp scaling-down of the present battleship standard. In any case, judging from recent experience, the "expert" element is by no means the final or decisive element in the moulding of American naval policy. —"The Engineer," April 13, 1928.

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## SOME THOUGHTS SUGGESTED BY THE ELECTRICALLY PROPELLED LINER *CALIFORNIA*.

By Capt. S. M. Robinson, U. S. N.

The steamship *California* has recently been completed and has just made her maiden voyage. Both events have been widely published in the technical and daily press. The ship has many interesting features and is number one, in many particulars, in the American Merchant Marine; but she really marks a distinct epoch in marine engineering and, more especially, the third epoch in the electric propulsion of ships.

The first epoch was marked by the U. S. S. *Jupiter* (now *Langley*) and the second by the U. S. S. *New Mexico*. In 1922, the author published a book entitled "Electric Propulsion." In Chapter I, which treats of the "History of Electric Propulsion and Types of Ships for which it is best adapted," the following summary of methods of propulsion was made:

### DIFFERENT TYPES OF MACHINERY.

"To recapitulate, the following table shows the type of machinery best adapted to each class of ship:

(1) Low speed cargo vessels of less than 3000 shaft horsepower—Diesel electric propulsion.

(2) Cargo and passenger vessels requiring more than 3000 shaft horsepower, but not enough to require the use of four screws—turbo electric propulsion.

(3) Passenger vessels of high speed and sufficient horsepower to require four screws—turbo electric propulsion.

(4) Destroyers, scout cruisers and light cruisers—geared turbines.
(5) Battleships and battle cruisers—turbo electric propulsion."

In this same chapter the further statement was made:

"It must be understood that the following subdivision of ships into classes can only be considered as a rough guide and not as an absolute one; also, the developments in the art are very rapid, and in a short time we may see a very large extension of the Diesel engine in the marine field over that laid down in the following subdivisions of classes of ships."

### POLITICAL AND NAVAL CONSIDERATIONS.

When this book was written, merchant ships were purposely divided into three classes, because it was believed that the development of the Diesel engine would make it applicable to vessels of "Class 2" and such has proved to be the case. Undoubtedly, a better division, at the present time, would be two classes, without reference to the number of screws. The first class would be propelled by Diesel engines and the second by electric propulsion. The dividing line would not be very exact, as it would depend on the type of service as well as speed and horsepower requirements, as will be shown later. The previous discussion was entirely along engineering lines, whereas, at the present time, there are political and naval considerations that must have a great influence in determining the types of propulsion for high-speed merchant vessels. To emphasize this point of view, it will be desirable to state briefly the principal characteristics of the *California*.

### FIRST LARGE COMMERCIAL VESSEL ELECTRICALLY PROPELLED.

The *California* was built by the Newport News Shipbuilding and Drydock Company, using propelling machinery furnished by the General Electric Company. She is designed for the Panama Pacific service of the International Mercantile Marine Company. She is the first large passenger vessel to be electrically propelled. In this respect she may be said to complete the development of electric ship propulsion; previously American marine engineers had introduced electric propulsion into almost every type of vessel, practically the sole exception being light, high speed war vessels. Electric propulsion, having had its inception and almost its entire development in this country, it is very gratifying to have the "capstone," so to speak, placed by American engineers.

### GENERAL DIMENSIONS.

The *California* is the largest passenger ship that has been built in this country. The general dimensions of the ship are as follows:

| | |
|---|---|
| Length overall, feet and inches | 601 3 |
| Length on waterline, feet | 595 |
| Length between perpendiculars, feet | 571 |
| Beam, molded, extreme, feet | 90 |
| Beam, molded, at promenade deck, feet | 77 |
| Depth, molded to shelter deck, at side, feet | 52 |
| Depth, molded to upper deck at side, feet and inches | 42 6 |
| Load draft, to bottom of bar keel, feet and inches | 32 3 |
| Displacement at load draft, tons | 30,250 |
| Deadweight capacity at load draft, tons | 15,300 |
| Gross tonnage, about, tons | 21,000 |
| Shaft horsepower, maximum | 17,000 |
| Shaft horsepower at cruising speed | 13,500 |

| | |
|---|---|
| Designed speed, knots | 18 |
| Cruising speed, knots | 16.5 |
| Cruising radius, miles | 15,400 |
| Cargo capacity, cubic feet | 478,700 |
| Refrigerated cargo capacity, cubic feet | 100,000 |
| Fuel capacity, tons | 5,349 |
| First class passengers | 384 |
| Tourist passengers | 363 |
| Crew | 350 |

If we consider carefully the figures given above, we shall see that here we have a vessel that would be useful as a naval auxiliary in time of war. It is true the speed is somewhat low but the basic requirement is met—that is, the ship is designated to operate, as a merchant vessel, at a lower rate of speed than she is capable of making, if she were to be converted to naval use. For cruiser work, it will, of course, be necessary to have vessels of much higher speed than the *California*, but certain large passenger vessels now are operated at 25 knots. In designing new vessels of this class, it will be a very simple matter to give them sufficient boiler power and, by the use of electric propulsion, sufficient horsepower to develop the high speed necessary for cruisers and commerce destroyers.

### POSSIBILITIES OF TURBO-ELECTRIC PROPULSION.

By the use of turbo-electric propulsion, this type of combination ship would really be much simpler than would appear at first thought. Considering first the boilers and oil-burning equipment, it is well known that merchant vessels do not burn oil at such high rates as are used by naval vessels at full power. For example, the *California* has boiler heating surface of 55,176 square feet and superheating surface of about 16,000 square feet and this is used to develop normally only 13,500 S. H. P. or a maximum of 17,000 S. H. P. This amount of heating surface could readily be made to develop three or four times this much horsepower and without any very radical changes. The most important changes would be in the oil-burning equipment, such as pumps, burners, heaters, etc. In the case of the propelling machinery, it is much simpler to provide additional horsepower by electric propulsion than by any other means. For example, by pole-changing, it will be possible to run the motors at any desired speed for merchant purposes and at any other desired speed for naval purposes, thus maintaining the same speed of turbine in both cases. Practically, it would probably work out best to speed up the turbine itself when using the vessel as a naval auxiliary, thus making it possible to get the best turbine design from the point of view of economy of fuel as a merchant vessel and economy of weight of turbine as a naval vessel.

### MERCHANT MARINE AND NAVAL REQUIREMENTS.

It is highly important that this type of ship should receive the study of our best engineers. At the present time, there is every indication that the people of this country are determined to have a merchant marine and an adequate navy. It is also being recognized that the two services are mutually dependent on each other. And most important of all, it is generally admitted that, if we are to have a merchant marine, it will require government assistance. The arguments now are not over whether there shall be government assistance, but how that assistance shall be given, whether in the form of direct government ownership, ship subsidies, or some other form of

government aid to private companies. In any case, where the government is contributing to shipbuilding, it will have something to say about the type of ship that is to be built and undoubtedly these ships will have to be built so as to contribute to the defense of the country in time of war. It is, therefore, time that American Naval Architects and Marine Engineers should give consideration to the Naval requirements of our new merchant marine.—"Marine Journal," April 1, 1928.

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## NEAR FUTURE DEVELOPMENTS IN NAVY AIRCRAFT ENGINES.

By Lieut. Comdr. J. M. Shoemaker, U. S. N.,
Head of the Engine Section, Bureau of Aeronautics, Navy Department.

Up to the present year, the efforts of air cooled engine manufacturers have been directed mainly toward attaining the utmost in reliability. Air cooled engine installations have inherent weight advantages, due not only to their construction, but also to the elimination of radiators, water lines, pump and water.

Although engine failures due to cooling system defects are automatically eliminated, it remained for the remarkable flights of 1927 to prove that air cooled engines are at least the equal of water cooled engines as regards reliability, and in all conditions of weather and temperature.

Having gained the confidence of the world in their products, American manufacturers of air cooled engines are taking steps toward refining and improving them. In addition to possible radical changes in engine design, improvement in present engines may be expected along the lines of stepping up the horsepower and reducing the specific fuel consumption. These improvements are possible due to:

(*a*) The soundness of design of the present engines, and their undoubted ability to stand higher loads.

(*b*) Availability of greatly improved aviation gasoline.

Paralleling the development of high-test motor fuels, the development of Ethyl Fluid, and its extensive use since 1925, has greatly improved the anti-knock qualities of domestic aviation gasoline. This has made possible the successful operation of modern engines on eastern (paraffin base) fuels whose natural anti-knock characteristics are extremely poor. If Ethyl Fluid could be used in any concentration, engine compression ratios might be raised to almost double that of present engines without regard to the natural anti-detonating qualities of domestic aviation gasoline. Unfortunately, however, excessive use of Ethyl Fluid has a very deleterious effect on valves and valve seats.

It has always been possible to write gasoline specifications which, by requiring a certain critical temperature of solution with analine, automatically shut out gasoline distilled from paraffin base crudes. This requirement, however, is not in conformity with the Federal Specifications Board specification for aviation gasoline. The Federal Specifications for gasoline as revised this year allow the military services to establish an acceptable list of approved aviation gasolines based on samples submitted for test. This will enable the Navy to establish a standard anti-knock value which must be equalled by any gasoline before it is placed on the acceptable list. Several large oil companies have been conducting extensive experiments with a view to producing a domestic aviation gasoline with high anti-knock characteristics, and these fuels will be ready to meet the Navy's requirements in the near future.

The means employed for increasing the power output of the Navy's present engines may be:

(*a*) Increase the R. P. M.,
(*b*) Increase the compression ratios,
(*c*) Supercharge.

The first step in this direction should be to increase R. P. M. and employ maximum allowable compression ratios with full throttle, sea-level operation. On top of this, engines may be supercharged to give rated power at full throttle at a predetermined altitude. This will necessitate the use of a throttle stop to limit throttle opening until a safe altitude is reached. The rotary induction system of fuel distribution now in use in modern radial engines lends itself beautifully to supercharging, as all that is required to obtain almost any desired manifold pressure at sea level is a change of impeller gear ratio.

The extent to which a service engine can be gradually stepped up in power is best exemplified by the British Bristol Jupiter engine series. All of these engines are nine-cylinder, static radial, air cooled, direct drive. The following table illustrates this:

| | *Jupiter Series IV* | *Jupiter Series VI* | *Mercury* | *Wright Cyclone* | *Pratt & Whitney Hornet* |
|---|---|---|---|---|---|
| Year | 1924 | 1926 | 1927 | 1927 | 1927 |
| Bore | 5.75″ | 5.75″ | 5.75 | 6.00 | 6.125 |
| Stroke | 7.5″ | 7.5″ | 6.5 | 6.875 | 6.375 |
| Piston Displacement, cu. in. | 1755 | 1755 | 1520 | 1753 | 1690 |
| R.P.M. | 1720 | 1870 | 2250 | 1900 | 1900 |
| Hp. | 430 | 480 520 | 870 | 540 | 538 |
| Compression Ratio | 4.9:1 | 5.3:1 6.3:1 | 8:1 (?) | 5.36:1 | 5.0:1 |
| Oil Consumption Lb./b.hp./hr. | .039 | .030 | ? | .025 | .025 |
| Oil used | Castrol | Castor | Castor | Mineral | Mineral |
| Fuel used | 20 % benzol | 30-38 % benzol | ? | D.A.G | D.A.G. + 2. c.c. Ethyl Fluid/Gal. |
| Weight, dry, lb. | 812 | 730 735 | 673 | 765 | 749 |
| Overall Diameter | 55″ | 53″ 53″ | 49″ | 54″ | 54.5″ |

The above table is a striking illustration of successive increases in the output of the same basic engine. The increase of power in the Jupiter VI over the Jupiter IV is due mainly to the higher compression ratio and higher R. P. M. The 6.3:1 compression ratio Jupiter VI is intended for altitude work. This engine incorporates a throttle stop, and full throttle is not used below 5000 feet altitude. The Bristol Mercury engine is a redesign of the Jupiter, with particular attention paid to reduction of diameter, and with a geared supercharger incorporated.

The two columns on the right give the general characteristics of the two large-displacement American radial engines. Both of these engines have built-in rotary induction systems. A comparison of the American engines with the British ones would seem to indicate that either the Cyclone or the Hornet is capable of having its output increased considerably by the means previously discussed.

It is worthy of note that the Jupiter engines require fuel with a high benzol equivalent, and must be lubricated with castor oil. American engine designers have been handicapped in comparison with the British by the insistence of the Navy (the Cyclone and Hornet were both designed for the

Bureau of Aeronautics) on the use of relatively low-test (paraffin base) aviation gasoline, and of mineral oil. As outlined earlier in this paper, expected improvement in the quality of fuel will remove the first of these restrictions. The use of mineral oil will still be required, however, in spite of the somewhat superior qualities of castor, because the supply of castor oil in quantity cannot be depended upon. In connection with a comparison of these British and American engines, the Jupiter engines require a separate oil cooler in the return line to the oil tank from the scavenger pump. The two American engines do not require oil coolers, thereby greatly simplifying engine installations.

It is noted from the table that the Mercury engine develops its rated power at 2250 R. P. M. This is entirely too high for good propeller efficiency except on airplanes of extremely high speed, for which the Mercury engine is intended. Raising the crankshaft speed is a comparatively simple means of obtaining more horsepower, however, provided the propeller shaft is geared down to a point which will give a good propeller efficiency with a reasonable propeller diameter. It is understood that the Jupiter Series VIII engine incorporates a 2:1 reduction gear, and this probably means that the crankshaft speed has been raised. Both the Hornet and the Cyclone have been experimentally fitted with reduction gears, which may become standard for these engines when used in comparatively slow, heavy duty planes.

The designers of the Hornet and the Cyclone concentrated on obtaining the desired power with engines of the utmost reliability and the minimum possible weight, and they brought out engines which, while extremely satisfactory as airplane power plants, do not lend themselves to streamlining as well as they might. The two means by which power plant resistance in a fuselage may be reduced irrespective of the type of cowling employed are:

(*a*) A reduction in engine frontal area;

(*b*) Increase in the longitudinal distance between the propeller hub and the maximum engine diameter.

The diameter of the Jupiter VI is two inches less than that of the Jupiter IV. This has been accomplished by shortening the cylinders, shortening the connecting rods, and reducing the diameter of the crankcase while retaining the same stroke. The diameter of the Mercury is four inches less than that of the Jupiter VI, which reduction is due to the one inch shorter stroke plus redesign of valve operating gear. It is probable that the diameters of the Hornet and Cyclone could be reduced in somewhat the same manner.

Redesigning the crankshaft of a radial engine to get the propeller hub farther in front of the engine would seem to present fewer difficulties than that of reduction in engine diameter. There is very little difference in this respect between the Jupiter IV and VI, but particular attention was paid to this feature in the design of the Mercury, in which the front cover of the crankcase was lengthened and tapered. Geared radial engines are automatically lengthened by the inclusion of the usual concentric gearing in the front crankcase cover.

The idea of building engines to fit airplanes, as discussed above, is one that will, it is hoped, influence American engine design during the next year. A further step, and a seemingly logical one, is to call upon engine manufacturers to supply with the engines a suitable exhaust manifold and suitable engine nose cowling. The design of both of these engine fittings is extremely important in assuring successful power plant operation, and it will be distinctly to the advantage of engine manufacturers to control such design. Heretofore in this country, both the engine cowling and the exhaust manifolding have been provided by the airplane manufacturers.

The results of attempts to cowl radial engines by airplane designers have been in many cases none too good. There is as much chance of undercowling an air cooled engine as of overcowling it, and in addition the cowling should be adjustable in the air for various temperature conditions. The engine manufacturers know the operating temperatures at which their engines run best, and how the cylinders can best be cowled.

The provision of exhaust manifolds by engine instead of airplane manufacturers should be a marked improvement, as engine manufacturers are vitally interested in insuring that the exhaust manifolds used with their product not only do not build up back pressure, but are an integral and serviceable part of the power plant assembly.

Radial air cooled engines are not necessarily the last word in aircraft power plants. They have, however, reached a high state of development. Their adoption for all Naval airplanes as well as their general use commercially is due to the widespread confidence in their reliability and ease of maintenance coupled with the great saving in power plant weight that they make possible. The soundness of design of present radial engines insures the possibility of increasing their power with no increase in weight and no decrease in reliability.

In conclusion, I would like to emphasize again the two thoughts that prompted this article. They are: "More power to the radial engine," and "Build engines to fit airplanes."—"Aviation," January 30, 1928.

## DIESEL-ENGINE FUEL INDICATORS.

It is one of the paradoxes of mechanical construction that of a number of engines built under exactly similar conditions and to an absolutely identical design, no two perform alike. Thus, of similar engines built, say, for a series of identical vessels, one engine on the test bed will give results superior to those of any other engine, while when installed on board and in actual service another will prove to be easily the most reliable. Various reasons can be advanced which may account, at least to a certain extent, for the wide discrepancies in the performances of similar engines, but it is not our purpose to enter into a discussion of these here; we are merely concerned with the statement of what is undoubtedly a fact. But just as the foregoing remarks apply to similar engines, so also they apply to similar parts of the same engine, and in particular they apply to the separate cylinders of a multi-cylinder Diesel engine. Here we have either six or eight absolutely identical cylinders fitted with pistons of identical design with fuel valves and the arrangements for admission of air and the exhaust of the products of combustion all of identical pattern, and yet, as every marine engineer who has had experience of the actual running of Diesel engines is aware, the performance of the various cylinders will differ between wide limits. This is shown not only by the size and shape of the indicator diagram, but also by temperature measurements of the exhaust gases or of the cooling water as it leaves the jackets.

This variation in performance reacts adversely not only on the actual running of the engine as a whole, but it also leads to various operating troubles and causes excessive wear and tear in one or more of the vital portions of the engine, thus conducing to excessive repair bills as well as to the possible laying up of the vessel while those repairs and replacements are being effected. It is obvious that if one cylinder, from no matter what cause, is not taking its share of the load, then the balance of the engine as a whole is upset, and in consequence excessive vibration may be set up or stresses may be induced in the frame or in the crankshaft which may have

serious results. Again, if one or more of the cylinders is not developing its correct proportion of the total power required for the propulsion of the vessel at the correct service speed, then the engineer-on-watch, in order to obtain the correct number of revolutions per minute of the main engine—this forming his criterion of the correct speed of the vessel—will increase the fuel supply to the engine as a whole, thus causing certain of the cylinders to develop more than their rated power, thereby leading to overheating of those cylinders, with the many evil consequences which follow in the train of overheating. The majority of Diesel engineers are fully aware of the consequences of the unequal working of the various cylinders of their engines, and they endeavor to watch for the development of such inequalities by frequent taking of indicator diagrams, correcting the same, where possible, by adjustment of the fuel-valve tappets or of the fuel pump itself. At the best, however, adjustments such as these are a matter of trial and error, and the necessity for them is only revealed at the time when the indicator diagrams are taken, say, once a week or even less frequently.

In order to give the engineer a continuous indication of the performance of each cylinder of his engine, the idea of fitting what have been termed "continuous fuel indicators" has been developed, and the idea appears to be gaining a wide measure of support among the users of marine Diesel engines. Briefly, the continuous fuel indicator is a device which shows at a glance the amount of fuel which is being supplied to each individual cylinder of the engine, and, at the same time, the apparatus can embody an arrangement of valves so that the supply to each cylinder can be varied at will. Thus, as soon as an inequality in the power developed by the various cylinders becomes apparent, by a simple adjustment of these fuel valves the engine can be "balanced" again until each cylinder is developing exactly the same amount of power. It necessarily follows that where a marked inequality continues to develop, steps should be taken to discover and to remedy the cause, but for what may be termed the inherent inequalities of the various cylinders the continuous fuel indicator should provide a means whereby these may be eliminated. It will be recognized that there are very many ways whereby the continuous indication of the amount of fuel supplied to each cylinder can be effected, and it is unnecessary to enter into a description of the various devices which are available here. It may be mentioned, however, that where these have been fitted to a marine Diesel engine and the inequalities of load on the various cylinders as revealed by the indicator have been corrected, then the uniformity of running of the engine has been very pronounced. And it should certainly follow that the uniformity of running thus obtained will be reflected in the greater reliability of the engine as a whole and in a substantial reduction in maintenance and upkeep charges.—"Shipbuilding and Shipping Record," February 9, 1928.

REFERENCES ON THE SUBJECT OF THE ABOVE EDITORIAL.

1. The Baulino Continuous Fuel Indicator—"The Engineer," February 11, 1927, p. 157. Describes the indicator installed on 2200 B. H. P., 400 R. P. M. Fiat engines built for Italian submarines.

2. Fuel Measurement for Individual Cylinders—"British Motorship," May, 1927, p. 51. Describes indicator installed on M. S. *Lalandia* fitted with Burmeister & Wain engines of 2500 I. H. P. at 130 R. P. M.

3. Mechanical Injection Oil Engines, by Oswald Wans—"British Motorship," June 1927, p. 113. Describes another method of measuring output of individual pump plungers.

## PRIMING AND IMPURITIES IN FEED WATER.

By Arthur W. Ewell,

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The experimental results described indicate that priming is increased by the presence of acids including hydrolized salts, non-dissociated substances such as sugar, slightly dissociated salts such as lead acetate, and insoluble oils. The increase in priming is attributed principally to increase in density and viscosity. Priming is decreased by insoluble solid particles and by highly dissociated salts. This decrease is attributed to the nucleii which such particles or ions furnish for the formation of vapor bubbles.

When water boils, steam forms in bubbles beneath the surface. When these bubbles rise to the surface and burst, a certain amount of water is carried with the steam into the steam space. This is termed "priming." If a superheater is used, this water is evaporated. There is little of published experimental data respecting the influence of impurities on "priming." Neither steam engineers nor steam engineering manuals nor textbooks are in agreement respecting the effect upon priming of the different foreign materials that may be present in feed water. The importance of this question, particularly in marine practice, where sea water is often employed, occasioned the investigation here outlined. After considerable preliminary experimentation the apparatus illustrated was adopted. The capacity of the boiling flask up to the stopper was 2150 cubic centimeters. A heating coil kept the temperature of the chamber in which the priming was investigated at about two degrees Centigrade below the boiling temperature. If the temperature had been kept at the boiling point for the particular day, a large amount of the water collected in the priming chamber might have been evaporated. If, for example, the current had been such as to give, without the passage of steam, a temperature slightly above the boiling point, the latent heat required for evaporation would have kept the temperature exactly at the boiling point and a large amount of priming water might have been evaporated without the experimenter's knowledge. Even though the temperature is kept two degrees below the boiling point, there will still be some evaporation. This, however, will introduce no error in comparative measurements since for the same time and the same jacket temperature the amount of evaporation will be the same. The heated jacket is essential, for otherwise a large amount of condensed steam will mask the relatively small amount of priming water. In these experiments there was practically no condensed steam to be considered.

The steam entered the top of the priming chamber and then passed through a plug of wire gauze which checked direct flow of the steam and retained any water carried along with the steam (priming water). The steam then escaped from the top of the other end of the chamber into the condenser. On account of evaporation from the graduate in which the condensed steam was collected, the amount read in the graduate was always less than the actual amount evaporated, as shown by the loss of weight of the boiling flask. At the close of a run the gauze was weighed together with a previously weighed wad of absorbent cotton which was used to collect the water, in the chamber, on the inside thermometer, on the stoppers and on the glass walls. Suitable precautions were taken during weighing, etc., to prevent evaporation. The thermometers were corrected for stem exposure and horizontal position. The following figures are illustrative of the readings in a typical experiment

DISTILLED WATER. BAROMETER 73.89 CM. HG.

| | | | | | | |
|---|---|---|---|---|---|---|
| Time | 4.29 | 4.31 | 4.33 | 4.35 | 4.37 | 4.39 |
| Outer Therm | 97.0 | 96.9 | 97.7 | 96.7 | 96.1 | 96.3 |
| Inner Therm | 99.40 | 99.85 | 99.95 | 100.05 | 100.03 | 100.10 |
| Graduate | 0 | 40 | 75 | 115 | 165 | 210 |

Water evaporated, 219 grams. Rate of evaporation, 21.9 grams per minute. Priming water collected, 1.87 grams. Priming per gram of water evaporated, 0.0085 gram = "Fractional Priming."

1 (*a*) Influence of common salt (sodium chloride):

Average values of fractional priming for fourteen alternate experiments were:

| Distilled water | 6 per cent salt solution |
|---|---|
| 0.0069 ± 0.0005 | 0.0053 ± 0.0005 |

(*b*) Strong salt solution—Average values of the fractional priming for six alternate experiments were:

| Distilled water | 14.85 per cent salt solution |
|---|---|
| 0.0053 | 0.0051 |

It is evident that the addition of common salt decreases the amount of priming for low concentrations (6 per cent). For higher concentrations the priming is about the same as that for pure water.

2. Influence of potassium chloride—Average results for five alternate experiments:

| Distilled water | 6 per cent KCl solution |
|---|---|
| 0.0055 | 0.0045 |

3. Influence of sugar—Six per cent solutions of sodium and potassium chloride are strongly dissociated. It appeared of interest to examine the effect of a dissolved substance which was not dissociated, and cane sugar was chosen. The following mean results were obtained for six alternate experiments:

| Distilled water | 5 per cent sugar solution |
|---|---|
| 0.0049 | 0.0056 |

The addition of sugar, therefore, increases the amount of priming.

Is there any explanation for this difference between the effects of sugar and salt?

When a bubble of steam forms at the bottom of a boiler, it overcomes three pressures: One is the pressure of the steam above the water, which is, of course, transmitted through the water; another is the pressure due to the depth of water in the boiler; and the third is the pressure resulting from the attraction between the molecules, which manifests itself as "surface tension." When a bubble of steam forms, it must spread apart the neighboring water molecules. Electrical charges reduce the surface tension. A familiar application of this effect is the "capillary electrometer," an instrument found in nearly every physical laboratory. The presence of electrical charges should, therefore, facilitate the formation of bubbles.

It is well known that such is the case in the formation of clouds. If electrified particles are introduced into a chamber filled with water vapor, a cloud results immediately, each electrified particle acting as a nucleus for the formation of a water drop. A similar effect would naturally be expected in the case of boiling. If we introduce electrified particles, they would serve as

nuclei and we would have a greater number of smaller bubbles, less violent boiling, and consequently less priming. This is what apparently happens in salt water. The majority of the molecules of salt in a weak solution break up into electrified particles, each molecule of salt giving an atom of sodium with a positive electrical charge and an atom of chlorine with a negative charge. We should find no such reduction in priming in a sugar solution, for sugar molecules do not break up into electrically charged particles.

The increase in priming with a sugar solution and lessened decrease in priming of salt water at high concentrations are undoubtedly due to the combined effect of greater weight of solutions compared with pure water, greater viscosity and slightly greater surface tension, all of which would tend toward more violent boiling and greater priming unless offset by some other factor, such as electrified particles.

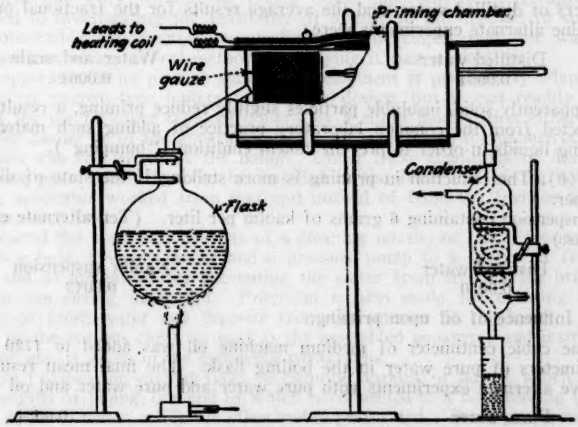
If the foregoing explanation is correct, a salt that gives few electrified particles should increase priming because it increases the density, viscosity and surface tension. A 6 per cent solution of lead acetate has a much smaller number of molecules than a similar solution of sodium and potassium chloride, and few of these molecules are dissociated into electrified particles.

4. Average fractional priming with

| Distilled water | 6 per cent lead acetate |
|---|---|
| 0.0040 | 0.0046 |

5. Bases such as sodium and potassium hydroxide give a large number of electrified particles in solution, but they also greatly increase the density and viscosity.

| Base | Mean fractional priming with Distilled water | 6 per cent solution |
|---|---|---|
| Sodium hydroxide | 0.0054 | 0.0053 |
| Potassium hydroxide | 0.0037 | 0.0037 |




*Sketch of apparatus for determining amount of priming*

The foregoing results with sodium hydroxide were obtained with normal boiling, the rate of evaporation being about the same as with the salts and sugar (approximately 20 grams per minute). The result for potassium hydroxide and the corresponding result for water apply to *very* gentle boiling—the rate of evaporation being about one-tenth the normal rate. When boiling approached the customary rate, violent foaming appeared above the potassium hydroxide solution, making observations very difficult, but there was no doubt that this excessive foaming was accompanied by abnormally great priming.

6. Influences of Acids.

| Acid | Mean fractional priming with Distilled water | 6 per cent solution |
|---|---|---|
| Sulphuric | 0.0046 | 0.0061 |
| Hydrochloric | 0.0050 | 0.0058 |
| Magnesium chloride | 0.0045 | 0.0069 |

Magnesium chloride is included because it largely hydrolyzes, giving a hydrochloric acid solution.

The foregoing solutions of strong acid are highly dissociated, resulting in a great number of electrified particles. One might conclude that the great increase in priming which they produce contradicts the foregoing explanation of the reduction in priming by weak solutions of sodium and potassium chloride. It should be remembered, however, that these acid solutions are peculiar. The large amount of heat liberated during solution indicates association between water and acid, and the boiling characteristics are entirely different from those of salt solutions, resembling more closely solutions of volatile liquids.

7. Influence of solid insoluble particles.

(*a*) 35 grams of powdered boiler scale were added to 1720 cubic centimeters of distilled water, and the average results for the fractional priming of nine alternate experiments were:

| Distilled water | Water and scale |
|---|---|
| 0.0064 | 0.0061 |

Apparently solid, insoluble particles slightly reduce priming, a result to be expected from the common laboratory practice of adding such material to boiling liquids in order to prevent violent ebulition ("bumping").

7 (*b*). The reduction in priming is more striking if the state of division is finer.

Suspension containing 6 grams of kaolin per liter. (Ten alternate experiments.)

| Distilled water | Kaolin suspension |
|---|---|
| 0.0050 | 0.0043 |

8. Influence of oil upon priming.

One cubic centimeter of medium machine oil was added to 1720 cubic centimeters of pure water in the boiling flask. The final mean results of twelve alternate experiments with pure water and pure water and oil were:

| | |
|---|---|
| Pure water | 0.0064 |
| Water and oil | 0.0068 |

This indicates that oil slightly increases the tendency to prime.—"Power," February 28, 1928.

# A HYDRAULIC METHOD OF CLEANING THE FIRE SIDE OF BOILER TUBES.

By H. A. Seiller, U. S. N.,
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Various methods and many appliances for removing the dirt, scale and other impurities that accumulate on the interior of boiler tubes and drums have been developed. As a result the watersides of water-tube boilers are, generally speaking, well preserved and the life of a boiler now is much longer than it was years ago.

On the other hand, practically the only device that has been satisfactorily developed for preventing the accumulation of soot on the tubes is the soot blower. However, this does not always keep the tubes and the spaces between them perfectly clean, and in time a certain amount of material, rich in sulphur and acids, may accumulate locally between the tubes. In most plants, afloat or ashore, the operating personnel remove this accumulation by routine cleaning after the boiler has steamed a specified number of hours, using special saws, lances, picks, etc.—a slow, laborious and not overly efficient method at the best. Some marine engineers, realizing the deficiencies of this method, periodically remove the boiler side casings to make the tubes more accessible and permit a thorough cleaning and removal of deposits.

A quick and efficient method of cleaning the firesides of tubes was recently developed by an officer of the United States Navy. It is a hydraulic method and consists of washing the tubes with hot fresh water under pressure. The method is an outgrowth of the keen competition in engineering performance among the ships of the Navy. The engineer officers, realizing the detrimental effect that dirty fire sides have on economy, spend much time and labor removing the soot, scale and other impurities, and the desire to accomplish this work without sacrificing other important items of upkeep work, led to investigating the feasibility of dissolving the deposits in some easily obtainable and inexpensive medium. It developed that hot water would answer, so a practical method of applying it was worked out.

The apparatus in its present state of development is particularly adapted for use on express-type boilers of Yarrow design, but it can readily be adapted for use on other types of boilers, by a slight modification either of the boiler casing or of the apparatus itself. In some types the unobstructed tube lanes run the length of the boiler. Under these conditions of design a cleaning panel would have to be installed in the front end of the boiler and the apparatus worked from that end instead of from the furnace side, as is described later.

In general the apparatus consists of a cleaning nozzle, or lance, connected through a hose, a water heater and a pressure pump to a source of fresh water, and of appliances for preventing the water from striking the brickwork or the casing insulation. Provision is also made for draining the mixture of fresh water and deposits from the interior of the boiler. An outline of the various parts, as well as the assembled apparatus and instructions for use, is given in the following paragraphs.

Figure 1 shows the various types of cleaning nozzles, or lances, which are short lengths of piping, one end of which is connected to a hose leading to a source of fresh water, while the other end is perforated. These perforations are of such size that their combined area is not greater than the cross-sectional area of the lance. This is necessary in order that the streams emitted will be under sufficient pressure to attack the deposits. As only a

small amount of water is used in cleaning a boiler, any water heater that will raise the temperature of 250 gallons of water 125 degrees F. in an hour can be used. One that can be readily constructed by the operating force is shown in Figure 2.

The combination drain trough and screen, Figure 3, consists of canvas cut to fit around the fire-side row of tubes and is fitted with side flaps to prevent water running over the edges. It has two drain holes for connections leading to the drain lines.

Figure 4 shows the drain trough connection to the lower drum of the boiler. It is made watertight by means of the packing bar, Figure 5, part 1; the channel bar, part 2; swing bolts, Figure 6, part 1; and tube clamps, part 2.

The packing bar is an ordinary piece of ¾-inch channel bar cut to suit the length of the lower drum of the boiler. The web of this bar is drilled to take 3/16-inch bolts, spaced six inches between centers. Soft rubber packing is placed between the flanges in such a manner that it extends beyond the flanges for a distance of at least a quarter of an inch. The other channel bar is of the same size as the packing bar, and holes are drilled in its web that will fair up with those drilled in the packing bar. The tube clamp is a clamp that has one end of each face curved to fit around the tubes of the boiler and the other end made flat to take the flat part of the swing bolt. The flanges of this clamp are drilled to take two ½-inch bolts, one of which is used to hold the swing bolt in place and the other to act as the pivot point of the swing bolt. The swing bolt is made in two pieces, one part being a bracket and the other a bolt. The flat end of the bracket fits between the flanges of the tube clamp, and the other end is drilled and tapped to take the bolt used for holding the packing bar against the drum.

In making the trough connection to the drum, the trough gasket connection is made first by placing the free end of the canvas between the webs of the channel and packing bars. Next, the bolts are installed in the web and then set up, making a watertight joint with the canvas acting as a gasket. The packing bar is then placed against the drum, the tube clamps and the swing bolts placed in position and the joint between the drum and the packing bar made by setting up on the swing bolts.

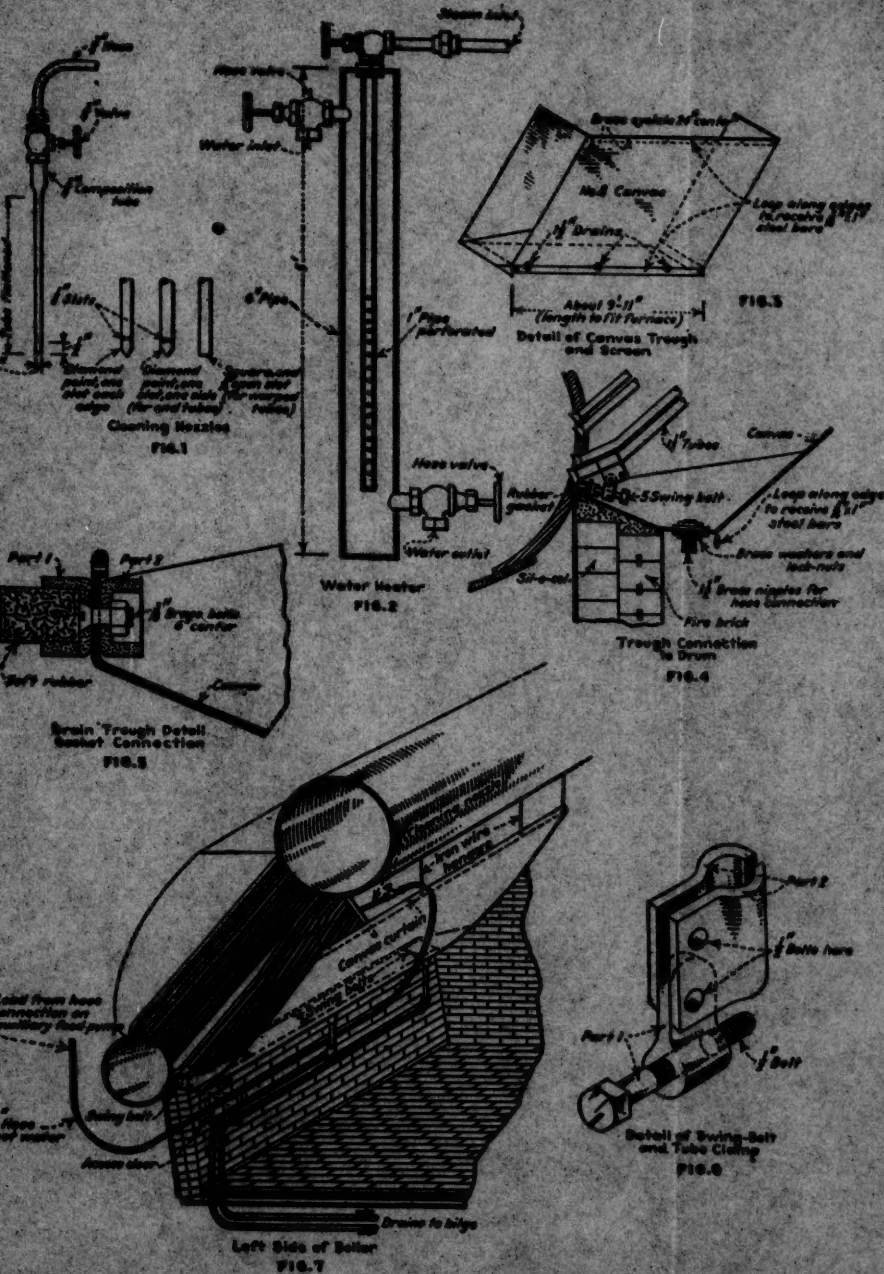
When this method is used, the apparatus is set up as shown in Figure 7. A staging is then rigged in the furnace of the boiler, and the operator stands on this and works over the top of the screen. The lance is placed in one of the tube lanes and the water is then turned on. After the lance has been moved back and forth several times, the water is shut off and the lance is shifted to another tube lane. The process is repeated until the water draining from the tube bank runs clear. The tubes are then blown with compressed air. After this is done, the apparatus is removed from the boiler and a slow fire lighted and maintained for a sufficient length of time to insure that all parts have been thoroughly dried.

### PRECAUTIONARY MEASURES.

The following precautions should be observed when this method of boiler cleaning is used.

1. The drum surface in the wake of the packing bar must be wire brushed and scraped and every effort made to obtain a watertight joint between the drum and the trough. Several inspections should be made while the cleaning is in progress to determine if any leaks have started. Any that do develop should be stopped immediately.

2. The nozzle valves, or lances, should be so selected that water will not be directed against the brickwork or the casing insulation.




DETAILS OF TUBE-WASHING APPARATUS.

3. A large piece of canvas should be placed across the row of tubes adjacent to the side casing of the boiler, or a screen similar to the one installed on the furnace side should be fitted.

4. Water must be delivered at a pressure of about 60 pounds, and the temperature of this water must not be allowed to drop below 165 degrees F. This temperature should be checked frequently while the cleaning is in progress.

5. When the cleaning of the tubes has been accomplished, blow out the tubes with compressed air.

6. After the tubes have been blown, dry out the boiler by means of a slow fire.

Originally, it was thought that the amount of water used would militate against the use of this method on board ship. However, it has been found that only 300 to 1000 gallons is required per boiler. The difference in the quantity of water used is accounted for by the condition of the boiler and the care with which the operators work. Furthermore, if the cleanings are so planned that they are accomplished at the same time that the water in the boiler is to be changed, the water that would ordinarily be dumped into the bilges can be used for cleaning the fire sides. Under these conditions the amount of water used is negligible.

This method has been given a thorough service test on many ships of the United States Navy, and it has been definitely demonstrated that three men can thoroughly clean a boiler in one day. Some ships have employed this method for about a year. No damage to the brickwork or the casing insulation has been reported, and the tubes have been kept free from dirt and scale. The increase in economy has been marked. Furthermore, not only are men who were used for cleaning available for other important work, but less fuel is burned and corrosion of fire sides of tubes has been prevented.—"Power," April 17, 1928.

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## OPERATING EXPERIENCES WITH 1300-POUND STEAM PRESSURE.*

By John Anderson.

### CORROSION PERIOD.

The failure of four boiler tubes only 330 hours after being cleaned, and with a very low condenser leakage, upset all previous deductions in regard to the prevention of scale troubles. It seemed impossible that less than 1/10 per cent condenser leakage would furnish enough solids to cause tube failure from scale formation. Continuous operation of the boiler would be impossible, unless some sort of feed water treatment could be used to prevent scale formation entirely. An inspection of the blisters showed them to be of a different nature than those which had occurred previously. The bulge was more pointed, and had a diameter of only ¾-inch. Moreover, the deposit inside the bulge was apparently of a different nature, being black in color and in larger quantities than before. This indicates the conditions observed after only 131 hours (5½ days) operation. Table VI gives analyses of this deposit found opposite the failures.

It will be noted that there is an excess of black iron oxide, obviously the product of electrolytic corrosion. In the first analysis a large percentage of scale-forming elements was present, though in the last 72 per cent was black iron oxide. From the analyses it was apparent that the cause of the

* Extract from a paper read before the Institute of Fuel on November 23, 1927.

trouble had changed from scale to corrosion. Whereas previously the solids in the raw water formed a protective coating over the tubes, now localized corrosion occurred, owing to the elimination of the raw water make-up without corresponding care in conditioning the boiler feed-water. The following is an analysis of boiler water taken during this period.

TABLE VI.—ANALYSES OF SCALE DEPOSITS IN TUBES.

| | Per cent | Per cent |
|---|---|---|
| Silica | 3.93 | 6.31 |
| Loss on ignition | 6.59 | ..... |
| Black iron oxide | 53.30 | 71.99 |
| Calcium oxide | 24 | ..... |
| Aluminium oxide | 12 | 7.06 |
| Calcium hydrate | ..... | 8.06 |
| Calcium sulphate | ..... | 1.22 |
| Copper oxide | ..... | 1.24 |
| Magnesium silicate | ..... | 4.15 |

TABLE VII.—ANALYSIS OF BOILER FEED-WATER.

| Date. | Gm. per gal. 7.1.27. |
|---|---|
| Calcium carbonate | 0.08 |
| Magnesium carbonate | trace |
| Sodium carbonate | 1.28 |
| Sodium sulphate | 0.15 |
| Sodium chloride | 0.58 |
| Iron oxide and alumina | 0.03 |
| Silica | 0.19 |
| Oil | 0.32 |
| Organic matter | 0.93 |
| Suspended matter | 1.11 |
| Total solids | 4.67 |

It will be noted that it is only very slightly alkaline and would probably become acid at times. Measurements taken during the next period indicated that there was 0.45 cubic centimeter per liter of oxygen in the boiler feed water. The three conditions of low raw-water make-up, possibly slightly acid boiler water, and relatively high oxygen in the feed-water, were therefore undoubtedly responsible for the corrosion.

Though all authorities are not in agreement as regards the role which oxygen plays in boiler tube corrosion, its elimination to a very large extent is apparently necessary. Hall explains its action by a laboratory experiment using a battery formed of two plates of boiler steel immersed in a low-sulphate water and separated by parchment paper. Carbon dioxide-free air is blown across the surface of one plate, thus causing an electrical potential between the two plates and inducing a current and metal flow from the air-free plate to the air-swept one. An ammeter indicates the current, and the amount of deposit of material on the air-swept plate can also be determined. Another experiment, illustrated in the Carnegie Institute of Technology Bulletin No. 24, shows the relative corrosion of drops of water containing measured concentrations of various salts, when kept on a boiler plate for stated intervals. Hall shows that the presence of sodium hydrate in a drop of water prevents corrosion, apparently by limiting the electrical

potential set up by differences in oxygen concentration on the inside and outside of the drops. These two experiments indicate the necessity of low aeration and the maintenance of soda alkalinity. The "Suggested Rules for Care of Power Boilers," published by the American Society of Mechanical Engineers, advise the maintenance of 255 parts per million of soda alkalinity.—"Engineering," January 13, 1928.

## THE BRINELL HARDNESS TEST AS APPLIED TO MARINE ENGINES.

In the Brinell hardness test a hardened steel ball is pressed into the smooth surface of the metal so as to make an indentation of a size such as can be conveniently measured under the microscope. The spherical area of the indentation being calculated, and the pressure being known, the stress per unit of area when the ball comes to rest is calculated, and the hardness number obtained. Within certain limits, the value obtained is independent of the size of the ball and of the amount of pressure, says A. B. Butterworth in "Marine Engineering."

While it is not claimed that the Brinell test will determine in advance the wearing qualities of metal, this article will show how any person supervising repairs to a vessel can determine with the least expense and loss of time whether liners, piston rings, follower plates, etc., are within certain tolerances as to hardness. In other words, the Brinell test should be used as a comparison test. The writer has had the opportunity of observing Brinell tests made on piston rings, new and old liners, etc., including those that have not given the best results, over a period of a year. The numbers ran from 116 to as high as 202; the average number found was 170 (these castings were made at various foundries). From these observations it is reasonable to expect good results with this test with small chances of any serious mistakes.

Quoting from one of the Bethlehem Steel Company's publications, "in soft grey iron castings the Brinell number will run from about 130 to around 175. The hard grey iron castings, which are machinable, will show Brinell numbers of from around 175 to about 250. The hardness represented by Brinell number 250 is about the limit of hardness for ordinary machine operation. Above this the casting is usually impossible to machine." Personally, I believe that a cylinder liner should be hard, not brittle, with a Brinell number around 200. The rings for this particular cylinder would be around 180, meaning that the rings would be softer than the liner in order to take up most of the wear. On a piston ring replacement for this particular cylinder a Brinell number as mentioned would be a requirement.

Under the present system, when a liner, piston follower, bull ring, etc., are installed in a cylinder at a shipyard, the castings must appear to be of good grade iron, free from blow-holes, flaws, porous places, cold shuts, scabs or other imperfections affecting their strength or appearance. This, however, requires only a superficial and often slighted inspection. Castings in shipyards are not as a general rule made to specifications. Even if they were, there are other things besides the composition of the metal that would make them unfit for installation and use in certain parts of a marine engine. The rate at which cooling takes place affects the characteristics of the casting almost as much as its chemical properties. Assume, however, that after this kind of inspection the liner is installed. If, after the engine has been operated a short time, we find the liner is very soft and is wearing down considerably, every practical operating engineer knows the results that follow.

### Steam Cylinder Liners.

| Silicon. Per Cent. | Sulphur. Per Cent. | Manganese. Per Cent. | Phosphorus. Per Cent. | Nickel. Per Cent. | Chromium. Per Cent. | Total Carbon. | Comb. Carbon. | Tr. S. Pound. | T. S. Pound. | Brinell No. |
|---|---|---|---|---|---|---|---|---|---|---|
| 2.60 | 0.065 | 0.62 | 0.310 | 0.14 | 0.21 | 3 436 | 0.580 | 3845 | 28,650 | 202 |
| 1.80 | 0.100 | 0.58 | 0.450 | 0.12 | 0.17 | 3.326 | 0.710 | 4161 | 33,620 | 202 |
| 1.34 | 0.086 | 0.60 | 0.110 | 0.15 | 0.16 | 3.436 | 0.710 | 4022 | 31,830 | 204 |
| 1.50 | 0.085 | 0.57 | 0.114 | 0.18 | 0.38 | 3 292 | 0.556 | 4492 | 37,410 | 201 |

### Piston Heads, Rings, Follower Plates (for above).

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 1.42 | 0.107 | 0.56 | 0.876 | 0.12 | 0.28 | 3.680 | 0.556 | 3126 | 21,890 | 185 |
| 1.67 | 0.103 | 0.62 | 0.402 | 0.20 | 0.35 | 3.708 | 0.492 | 3190 | 21,090 | 185 |
| 2.45 | 0.071 | 0.60 | 0.106 | 0.20 | 0.19 | 3.436 | 0.682 | 3853 | 27,850 | 185 |
| 1.87 | 0.125 | 0.58 | 0.800 | 0.15 | 0.15 | 3.572 | 0.682 | 3040 | 23,780 | 185 |

### Diesel Engine Liners.

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 1.15 | 0.083 | 0 56 | 0.140 | 0.14 | 0.32 | 3.108 | 0.764 | 4733 | 41,780 | 229 |
| 1.57 | 0.112 | 0.58 | 0.414 | 0.12 | 0.16 | 2.970 | 0.680 | 4741 | 44,570 | 229 |
| 1.69 | 0.102 | 0.55 | 0.720 | 0.28 | 0.23 | 3.272 | 0.874 | 4400 | 37,200 | 229 |
| 2.48 | 0.100 | 0.54 | 0.438 | 0 23 | 0.18 | 3.064 | 0.572 | 4777 | 39,195 | 229 |

### Pistons and Rings for Above.

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 2.35 | 0.111 | 0.57 | 0.880 | 0.10 | 0 16 | 3.272 | 0.736 | 3642 | 31,040 | 217 |
| 1.12 | 0.094 | 0.56 | 0.096 | 0.14 | 0.15 | 2 972 | 0.736 | 4550 | 41,780 | 217 |
| 1.04 | 0.095 | 0 59 | 0.470 | 0 30 | 0.26 | 3.598 | 0.880 | 3990 | 37,880 | 217 |
| 1.60 | 0.125 | 0.57 | 0.828 | 0.20 | 0.25 | 3.436 | 0 710 | 3785 | 27,060 | 217 |

As the rings supplied to vessels according to the above requirements may be hard or soft and spongy, it cannot be expected that all the rings will give the best results. As an illustration: On one occasion, when a set of intermediate piston rings was furnished a vessel, we were instructed to be careful in hauling them from the shop to the vessel. We decided that if they were too tender to be hauled, they certainly were not fit to be placed in a vessel's cylinder, and upon arrival their fate was quickly decided. The shop ordered out another set (no Brinell number was taken), and upon completion these were installed in the cylinder. During a voyage, when the vessel had to compound on account of high-pressure cylinder trouble and the load on the intermediate cylinder was increased, the rings cut the cylinder to such an extent as to make it necessary to renew the liner. The adoption of a Brinell number and its use on these particular rings would probably have prevented this trouble and expense.

It must be understood that to adopt Brinell hardness numbers for piston rings and the like, when the hardness of the liner is not known, is practically useless. To accomplish the best results it is recommended that when a new liner is installed the hardness number should be obtained, top and bottom, at four different positions. The liner could then be marked in a conspicuous place with the proper number. As far as we know there is no portable hardness testing machine on the market, but it is hoped that some farsighted manufacturer will see the opportunity for a light, compact, reliable, portable testing machine, for the purpose of determining the hardness of liners, etc., previously installed. When this is done a great deal of cylinder trouble will be avoided.

A new high-pressure cylinder liner was installed July 23, 1923, on a vessel's triple-expansion engine, the bore at that time being 18½ inches. The liner was rebored on April 11, 1924, and again on January 13, 1925. The liner was finally removed on October 12, 1925, the bore at that time being 19½ inches. Six or more sets of piston rings were worn out in this time, the reason being the extreme softness of the liner, which had a Brinell number of 116. All of the worn particles of metal passed through the cylinders, valves, and packing, clogging up the cylinder cocks, etc. Although it would be impossible to determine the cost of wear and tear on the engine in this particular case, nevertheless it is obvious that this expense could be very materially reduced, if not entirely eliminated.

While it is commendable and economical to conserve scrap metal, nevertheless it should be remembered that scrap metal has been a contributing factor to the inferior grades of castings that are furnished in vessel replacements. Under no consideration nor at any price should inferior castings be used for cylinder liners, piston heads, piston rings, follower plates, etc. Repair yards do not take proper precautions in the mixture of iron castings, and the result is that anything will go into the furnace. Repair yards are not fully to blame. Vessel operators can eliminate this trouble entirely in demanding a better grade of castings and get the most for their money. Sooner or later somebody must work out the problem, and in order to get good results, both the representatives of the operators and the officials of the repair yard must co-operate. It is an injustice to both to expect a repair supervisor to look at a casting and judge whether it is up to standard or not. All of us attempt to do this because it is common practice and there are no better methods available.

When it is considered that thousands of dollars are expended annually for vessel repairs and replacement, it is remarkable how little attention is given to making sure that proper materials are being used. Of all the large operating companies I cannot recall any that specify the composition of the castings and follow them up with any kind of a test to see that they are up

to specifications. This is probably the reason that we often hear the expression: "We cannot make as good grade of casting in this country as they do in Europe." Of course we can, if they are demanded and followed up with sufficient tests.

Two cylinder liners were replaced in a motor vessel's engine of American make. These liners had run 241,977 miles in five years. Test pieces were taken off the extreme ends of one liner. The Brinell number on top showed 170-174, and on the bottom end 179-183. The chemical analysis showed that these castings were of the ordinary grade used in steam-engine cylinders. The writer is of the opinion that these liners should have been harder, especially for this class of engines. As Diesel engines are running steam a close race, it seems only logical to require exacting specifications and a hardness test for cylinder liners, pistons, rings, etc., on this class of vessel.

Take a typical automobile cylinder casting, analyzing silicon 2.660 per cent, phosphorus 0.077 per cent, sulphur 0.145 per cent, manganese 0.600 per cent, total carbon 3.020 per cent, tensile strength 36.740 pounds per square inch, Brinell hardness 223 to 229. These castings machine readily, but it is very rare indeed to secure a casting such as this in marine repairs. One automobile company checked up a cylinder on one of its test cars after the car had run 45,000 miles. The wear in the cylinder was found to be less than five ten-thousandths of an inch. The reason was plain—the proper material was in the cylinder. Although the automobile industry is a comparatively young industry, we can profit by their methods and requirements in the use of metals. One of the greatest assets and advertisements of the American-built automobile today is the fact that nothing but the best material is used in the structural parts.

While the scope of this article and the data which it contains are limited, it is hoped that the article will bring out discussion that will eventually show from a dollar and cents point of view that material tests on vessel replacement work will pay.

The above compositions were selected principally for their hardness as shown by the Brinell numbers in the last column.—"Mechanical World," April 13, 1928.

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## DIRECT CORROSION TEST.

The following is an extract from an article written by Mr. John C. Jennings, chief chemist of the W. B. Dick & Co., Ltd., in connection with investigations made by Mr. Horace J. Young, F. I. C., a copy of whose paper appears on page 149 in the Notes of the February issue of this JOURNAL.

A modified direct-corrosion apparatus was devised for the sole purpose of examining the results along the lines used by Mr. Young. In every case the results obtained were in accordance with expectations.

The sample of used oil from the crankcase of a Diesel engine was contaminated with a considerable amount of water containing dissolved mineral salts. It had not been subjected to any cleaning process. Ordinary laboratory examination showed it to be in a condition to cause corrosion. A portion of the sample was cleaned by means of a laboratory centrifugal machine. It was then tested and gave no signs of corrosion, confirming that the oil itself was still in good condition and could be used with perfect safety for lubricating purposes.

The work clearly shows that corrosive effects can in no case be due to good lubricating oils. If these do occur they are due not to the development of inherently bad properties dormant in the oils themselves, but to the

TABLE 5 (TYPICAL DIRECT-CORROSION EXPERIMENTS).

| Oil Used. | Atmosphere. | Examination of Test Plate. |
|---|---|---|
| Oil "A" oxidized with dry air. | Dry air. | No sign of pitting or corrosion on steel or white metal after 100 hours. |
| Oil "A" oxidized with dry air. | Products of combustion from fuel oil. | Sign of corrosion clearly visible to the naked eye on steel and white metal after 12 hours. |
| Oil "A" oxidized with dry air. | Air saturated with moisture. | Very slight pitting of steel plate after 80 hours. White metal unaffected. |
| As above + 0.3 per cent water. | Ordinary air. | Corrosion visible on steel and white metal after 24 hours. |
| As above + 0.3 per cent sea water. | Ordinary air. | Corrosion visible on steel and white metal after 10 hours. |
| Used oil containing moisture and sludge from crankcase of Diesel engine. | Ordinary air. | Excessive corrosion visible after 10 hours. |
| Same oil cleaned and dried. | Ordinary air. | No corrosion after 100 hours. |

acquisition of contaminants from external sources. The real problem, therefore, resolves itself into an attempt, preferably, of altogether preventing this contamination, or if this is not entirely practicable, of minimizing its effects as far as possible. The first is a matter for the Diesel engine designer, and it is well known to engineers that certain types of Diesel engines built at present are notorious for the quantity of water and fumes, which are able to leak into the crankcase. It must be borne in mind that condensation of water vapor also takes place when the engine cools. This condensate varies with the conditions, but in certain circumstances may be sufficiently great to cause trouble. The other aspect of the problem is one in which users and manufacturers of lubricating oils may work together.

### SOME CONSIDERATIONS FOR THE ENGINEER.

The points which require most careful attention are:

(*a*) The quality of the fuel oil used must receive more consideration than it has done in the past, to ensure that it cannot cause any harmful effects.

(*b*) Of equal importance is the choice of a suitable lubricating oil. This oil must not only meet the requirements as regards actual lubrication, but must also have the properties of resistance to oxidation and emulsification. The tendency to form sludge and acidity must be as little as possible, and the

demulsibility value not only in the unused oil, but also in the oil during service, must be the maximum obtainable. These features are interdependent, because the formation of the sludge and acidity has the effect of materially decreasing the demulsibility value of the oil. Oils which give high sludge values after comparatively short periods of working result in persistent emulsification. The emulsion so produced will hold sludge, water, and other impurities, which are subsequently carried round in the circulating system. It will be readily understood that inefficient lubrication, and possibly corrosion with all its attendant troubles, is inevitable. It may, therefore, be wise to point out that too much reliance must not be placed upon the demulsibility value of the new oil. Its ability to retain this value during service is important.

(*c*) Having obtained a good lubricating oil it is, above all things, necessary to keep the system as clean as possible. This not only includes the engine itself, but also the storage tanks and, particularly, the cleaning apparatus. The intake air to the engine should also receive attention and should be as dry and clean as possible. Even, however, with these precautions it will be impossible in the case of many of the engines in use at the present time to keep the lubricating oil free from contamination, and an efficient cleaning process is absolutely essential. This again emphasizes the importance of the consideration given in (*b*), for an oil which develops a poor demulsibility value in use will render the cleaning process extremely difficult and wasteful. The oil will be very quickly vitiated until it becomes apparent that initial costs are of secondary importance, and that such an oil is not only a very serious disadvantage to the engine, but an extremely expensive one expressed in terms of actual oil costs.

It will be realized from the paper that most contaminants likely to have corrosive properties will be associated with products having a gravity of 1.00 or over; consequently, there will be a tendency for separation to take place by gravitation. This is a process which may go on in the crankcase itself, particularly when the engine is shut down, and which certainly does take place in the settling tanks, the extent varying with the quality of the oil. It will thus be clearly evident that it is easily possible to run off a sample from the crankcase or settling tank, which would show a concentration of these undesirable contaminants, whereas the amount in the main body of the oil may be insufficient to cause any harmful effects. No reliance can therefore be placed on any laboratory test unless the complete history is known of the samples tested. The extreme importance of submitting samples for analysis which are truly representative of the oil in actual circulation will be realized.

To sum up, the engineer, if he wishes to obtain maximum efficiency and avoid corrosive troubles, must insist upon a first-class oil, and must make it his business to keep it clean. A good oil can never develop corrosive properties during normal use; it can only acquire them through contamination.

The author wishes to express his appreciation of the work carried out by Mr. H. W. Harrison, B. Sc., of the Research Department of W. B. Dick & Co., Ltd.—"The Marine Engineer and Motorship Builder," April, 1928.

# ASSOCIATION NOTES.

## ANNUAL BANQUET.

The annual banquet of the Society held in Washington D. C., on March 29, 1928, was one of the most successful and well attended events ever held by the Society. Detailed account of special features will be found elsewhere in this issue.

## PRIZE ESSAY.

Attention is invited to the Special Notice on the back cover of this issue relative to the Prize offered for the best article submitted during the year. Rules governing the award of this prize have been materially changed. Under the revised rules all articles received this calendar year will be considered in awarding the prize, including the articles that have already been submitted.

## WORLD ENGINEERING CONGRESS, TOKIO, JAPAN, OCTOBER, 1929.

Through the Department of State, the American Society of Naval Engineers, has been invited to participate in the World Engineering Congress in Tokio in the fall of 1929. Rear Admiral Halligan, Engineer in Chief, U. S. Navy, the President of the Society, is a member of the American Committee. The Society has appointed Rear Admiral C. W. Dyson, U. S. Navy, Retired, as chairman of a committee to represent the American Society of Naval Engineers, and to work with the American Committee. The membership of the committee will be announced later.

The committee invites suggestions from our members as to suitable topics and authors for papers. Below are given the preliminary announcement and a tentative arrangement of the subjects to be discussed.

## WORLD ENGINEERING CONGRESS IN TOKIO
## OCTOBER, 1929.
### PRELIMINARY ANNOUNCEMENT.

The Kôgakkai (Engineering Society of Japan) announces hereby a World Engineering Congress to be held at Tokio, for two weeks, toward the end of October, 1929, and cordially invites the participation therein of Governmental Departments, Universities, Institutes, Associations and individuals, collaborating in Engineering.

The Congress proposes to discuss various engineering subjects (tentatively arranged as annexed herewith) in anticipation eventually to initiate and promote the international co-operation in the study of engineering science and problems in all its branches and to cultivate a feeling of brotherhood among engineers of the world.

More detailed announcements will be issued later. Correspondence and inquiries should be addressed in the meantime to the Kôgakkai, Marunouchi Building, Tokyo.

BARON K. FURUICHI,
*President of the Kôgakkai.*

August, 1927.

1.—General problems concerning Engineering:
Education, Administration, Management, Statistics, Standardization, International Co-operation of Engineers, etc.

2.—Engineering Science:
Strength of Materials, Thermodynamics, and other Scientific Researches.

3.—Public Works:
Railways, Highways, Harbour Engineering, River Engineering, Canals, Municipal Engineering.

4.—Communication and Transportation:
Ocean and Inland Navigation, Aereal Navigation, Telegraph, Telephone, etc.

5.—Power:
Resources, Production and Distribution.

6.—Architecture and Structural Engineering.

7.—Mechanical and Electrical Engineering.

8.—Chemical Industry.

9.—Textile Industry.
10.—Shipbuilding and Marine Engineering.
11.—Aeronautical and Automotive Engineering, Road Vehicles.
12.—Mining and Metallurgy.
13.—Engineering Materials.
14.—Fuel (solid, liquid and gaseous) and Combustion Engineering.
15.—Water Works, Drainage, Heating and Ventilation, Illumination, Refrigeration.
16.—Scientific Management.
17.—Miscellaneous.

### LOUVAIN MEMORIAL.

The Chairman, Mr. Edward Dean Adams, of the Committee on the Louvain War Memorial to American Engineers, in his last financial statement, announced that of the $83,000 necessary for the Memorial, only $62,000 have been contributed. This amount has been given by only three and one-half per cent of the members of the participating societies. Inasmuch as the other payments are due soon, and the final payment will be due in August he urges those who desire to contribute, to do so at an early date.

The Memorial is to be dedicated on the Fourth of July of this year.

The Committee has received to date the names of more than 900 Engineers of the United States who gave their lives in the Great War 1914-1918.

### ADDRESSES.

Particular attention of Naval Members is invited to the fact that hereafter changes in address will be made only upon receipt of request. It is no longer practicable to ensure delivery of your JOURNAL by making changes from daily "Orders to Officers."

### MEMBERSHIP.

The following have joined the Society since the publication of the previous issue of the JOURNAL:

NAVAL.

Alexander, Boyd R., Lieutenant Commander, U. S. N.

Braswell, M. T., Ensign, U. S. Coast Guard.

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CIVIL.

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Stine, Wilmer E., 611 Fairview Avenue, Brooklyn, N. Y., from Associate to Civil Membership.

## ERRATA.

Page 283 (May 1928) paragraph 5, line 4, the word "authorize" should be "amortize".